VOL-IV NO-2

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

S·A·E

FEBRUARY 1919

SOCIETY WINTER MEETING
VICTORY DINNER
NEW YORK
FEBRUARY 4–6

SOCIETY OF AUTOMOTIVE ENGINEERS INC.
29 WEST 39TH STREET NEW YORK

Criss-Crossed in every direction

NOTE the interwoven or cloth-like appearance of the structure in this micro-photograph. By reason of the exclusive process employed in alloying Non-Gran Bronze, this fibrous, crystalline structure results, requiring a greater degree or duration of frictional pull to produce destructive wear.

Our entire production resources are devoted to this one alloy — Non-Gran. Production is maintained under the inflexible policy of *one formula, one method, one result.*

This is a factor of vital importance in the selection of a bearing bronze. Even though the bronze you are using *today* may be satisfactory, what assurance have you that *tomorrow's* shipment will be likewise, if produced by a foundry making a number of *different* alloys?

The present era is one of specialization. Non-Gran is produced by specialists. Its unvarying dependability is the same today as years ago, and the same tomorrow as years hence.

In our new plant the latest devices and equipment have been installed to further reduce the element of human fallibility in maintaining the uniformity of Non-Gran Bronze.

The term "Non-Gran" symbolizes

A reputation made by quality maintained by uniformity

HIGH SPEED
NON-GRAN
BEARING BRONZE

American Bronze Corporation

Berwyn Pennsylvania

THE JOURNAL OF THE SOCIETY OF AUTOMOTIVE ENGINEERS

Vol. IV February, 1919 No. 2

ANNUAL MEETING

NEW YORK, FEBRUARY 4-6

PROGRAM

GENERAL SESSION

Wednesday Morning, Feb. 5

Tanks. Lieut.-Col. Herbert W. Alden, U. S. A.
Automotive Ordnance Apparatus. Lieut.-Col. W. G. Wall, U. S. A.
Principles of the Wheeled Farm Tractor. Edward R. Hewitt.
(Printed in this issue of THE JOURNAL.)
Automotive Applications of Marine Engines in the War. George F. Crouch.
(Business and Standards Committee matters will be included in this session)

AUTOMOBILE SESSION

Wednesday Afternoon, Feb. 5

Probable Effect of Aeronautic Experience on Automobile Practice.
A Symposium by Henry M. Crane, Howard Marmon and O. E. Hunt.
High-Efficiency Automobile Engines. D. McCall White.
The Story of the United States Standard Truck. J. G. Utz.

FUEL SESSION

Thursday Morning, Feb. 6

More Efficient Utilization of Fuel. Chas. F. Kettering.
Unmined Supply of Petroleum in the United States. Dr. David White.
Status of Refinery Practice in the United States. Dr. E. W. Dean.
An Interpretation of the Engine Fuel Problem. Dr. Joseph E. Pogue.
Status of Engine Efficiency in the United States. Dr. H. C. Dickinson.
Mexico as a Source of Petroleum and Its Products. E. De Golyer.
(Printed in this issue of THE JOURNAL)

AERONAUTIC SESSION

Thursday Afternoon, Feb. 6

Fixed Radial Cylinder Engines. John W. Smith.
(Printed in the January issue of THE JOURNAL)
Proportioning Plane to Engine. Lieut. Alexander Klemin.
Problems of the Naval Aircraft Factory. Commander F. G. Coburn.
Airplane and Seaplane Engineering. Commander H. C. Richardson.
Navy Dirigibles. Starr Truscott.
Operation of Naval Aircraft. Commander J. H. Towers.
The Liberty Aircraft Engine. Jesse G. Vincent.

Making the Airplane a Utility. Grover C. Loening.

Forest Products for Aircraft Use. Clyde H. Teesdale.

The sessions listed above and the Standards Committee meeting will be held in the Engineering Societies Building, 29 West Thirty-ninth Street.

STANDARDS COMMITTEE MEETING

Tuesday Morning, Feb. 4

REPORTS

BALL AND ROLLER BEARINGS DIVISION
- Annular Ball Bearings, Extra Small Series.
- Annular Ball Bearings, Extra Large, Extra Light Series.
- Annular Ball Bearings, Extra Large, Light Series.
- Annular Ball Bearings, Extra Large, Medium Series.
- Angular Contact Ball Bearings.
- Separable Type Annular Ball Bearings, Extra Light Series.

CHAIN DIVISION
- Roller Chain Nomenclature.

ELECTRICAL EQUIPMENT DIVISION
- Cable Terminals for Ignition Distributors, Generators, Meters and Switches.
- Sleeve-Type Starting Motor Mountings.
- Non-Magnetic Magneto Shims.

ENGINE DIVISION
- Engine Support Arms.
- Magneto Couplings, Flexible Disk.

IRON AND STEEL DIVISION
- Steel Castings Specifications.

MISCELLANEOUS DIVISION
- Nuts for Machine Screws.
- Flexible Metal Tubing.

SPRINGS DIVISION
- Spring-Bolts for Commercial Cars.
- Spring-Bolts for Passenger Cars.
- Spring-Pins for Commercial Cars.
- Method of Greasing and Oiling Spring-Bolts.
- Leaf Points.
- Leaf-Point Nomenclature.
- Tests for Parallelism.
- Eye-Bushing and Bolt Tolerances.
- Specifications for Leaf Springs.
- Spring Widths, Lengths, Eye and Clip Diameters for Commercial Cars.
- Spring Widths, Lengths, Eye and Clip Diameters for Passenger Cars.
- Rebound Clips, Spacers and Bolts.
- Wrapped Eyes.
- Width of Spring Ends.
- Width of Spring Brackets.

TIRE AND RIM DIVISION
- Solid Tire Sizes.
- Base Bands for Solid Tires.
- Industrial Truck Tires.
- Base Bands for Industrial Truck Tires.
- Allowable Tolerances for Felloe Bands.
- Edges of Felloe Bands.
- Wood Felloe Dimensions for Pneumatic Tire Rims.
- Pneumatic Tires for Motorcycles.
- Wood Spokes for Pasenger Car Wheels.
- Valve Hole Sizes for Automobile Rims.
- Solid Tire Sections.

LADIES' NIGHT

ENTERTAINMENT, RECEPTION AND DANCE

NORTH BALLROOM, HOTEL ASTOR, 8:30 WEDNESDAY EVENING, FEB. 5

Promises to be one of the most popular and successful events of the meeting. The reception will be followed by a dance, and diversion of interest to those who do not dance. Admission will be by ticket for which there is no charge. Tickets will be reserved for members making application on the forms which have been mailed to them and will be delivered at the registration desk during the meeting on Feb. 5.

VICTORY DINNER

GRAND BALLROOM, HOTEL ASTOR, 7 O'CLOCK, THURSDAY EVENING, FEB. 6

The Grand Ballroom at the Hotel Astor seats a much larger number than could be accommodated before on similar occasions, but members are urged to make immediate reservations with check for the number of covers desired. Tickets are $6.00 each. Seats will, so far as possible, be assigned in order of receipt of remittance.

Ladies will not attend the dinner, but those who care to hear the speeches will be admitted to the balcony.

THE MIDNIGHT WHIRL

CENTURY GROVE, THURSDAY EVENING, FEB. 6

DANCING FROM 10:30 TO 11:30. PERFORMANCE AT 11:30

Tickets are $2.50 per person, including war tax. The Committee will assign seats as equitably as possible. There will be dancing also during the intermission and following the show.

A Theory of Plate Springs

By David Landau* and Percy H. Parr

Illustrated with Drawings

This paper is intended to give a synopsis of the theory of plate springs developed by the authors and to indicate some of the results obtained by the application of the theory to automobile springs. The complete mathematical analysis appears in the April and December, 1918, issues of the *Journal of the Franklin Institute* and this paper is largely an abstract from the manuscript prepared for that Journal. A second installment of the paper will appear in an early issue.

The study of the particular problem with which the present paper is concerned was commenced by us nearly sixteen years ago; the research was carried on intermittently, and some of the fundamental relations were discovered but a few years ago. During the last ten years we have searched the technical literature of the world to discover what researches, if any, had previously been made by the earlier mathematicians and engineers which resembled our own. We have found but a single paper of importance, that of the famous French engineer, M. Philips, who made a study of the leaf spring initiated on lines akin to our own; his theses, however, gave the mathematical relations in such complex form as to be of but slight use to the engineer and designer. The theory of Philips was not developed to an extent comparable with our researches.

The application of the results of our investigation to practice has resulted in springs having an endurance, or life, far greater than that usually obtained.

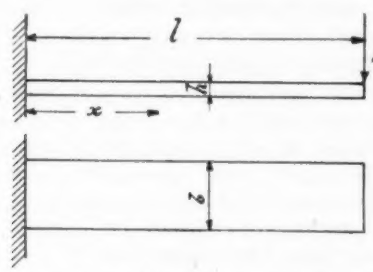


FIG. 1—THEORETICAL SPRING LEAF

To lead to a complete understanding of our theory of the leaf spring it seems well to explain first, in as elementary a manner as possible, the commonly accepted theory.

The Ordinary Theory

Consider a single leaf as shown in Fig. 1, having a length l, a constant thickness h, and a constant width b; fixed at one end and subjected to a load P at the other end. Under these conditions the plate will deflect a certain distance, say d, as shown in Fig. 2. Of course, we assume in this exposition that any leaf spring may be considered as a beam encastrè at one end and loaded at the other, as is the case with a cantilever spring; other forms, such as semi-elliptic springs, may, if desired, be considered as beams supported at the ends and loaded at the center; the final result, however, is exactly the same as if they are considered as two cantilevers joined at the points of encastrèment. It suffices, therefore, to consider the case of the cantilever.

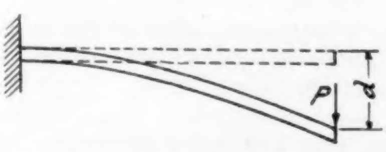


FIG. 2—EXAMPLE OF DEFLECTION PRODUCED BY LOAD

Suppose, now, that we arrange some suitable mechanism which shall vibrate this simple one-leaf spring through the distance d such a number of times as shall cause it to break. Clearly, the fracture will occur at the point of fixation, for the bending moment, which is evidently $P(l - x)$, there attains its maximum value Pl; and, as the section of the leaf is uniform, the stress is a maximum at the same section as the bending moment.

The number of vibrations required to break the one-leaf spring having been determined, we may now propose the problem of constructing a spring to carry twice the load, or $2P$, and to deflect the same distance d, and also to withstand the same number of vibrations as before. How should we, in accordance with the present-day theory, make a spring to fulfill these requirements, it being understood that the length l must not be altered? The answer of the present-day theory is given by Fig. 3.

A second leaf, called the short leaf, and in this case having a length $\frac{1}{2}l$, would be placed under the original leaf. The prevailing theory advises that the resulting spring of Fig. 3 will support, with equal safety and equal deflection, a load double that of our original one-leaf spring, or $2P$, if the load for the single leaf is P.

Again, the theory in common use today indicates that the spring to carry $3P$ must be made as shown in Fig. 4, where it will be seen that the second leaf has a length of $\frac{2}{3}l$ and the short leaf a length of $\frac{1}{3}l$. In general, this theory assumes that the number of leaves is directly proportional to the load. It should be noted that the "steps" or "overhangs" are all equal.

FIG. 3—THEORETICAL TWO-LEAF SPRING

For the present we are considering only springs composed of leaves of the same cross-section; later on we shall discuss the effect of varying the sections, or "grading" the springs.

*Consulting Engineer, New York City.

Now, suppose that we consider a spring such as that shown in Fig. 4, and ask if the stresses are the same in all of the leaves, and if the maximum stress in each leaf is the same as that in the simple leaf of Fig. 1. The answer given by those who expound the ordinary theory is about as follows:

In the case of Fig. 3, they say that, since the load is doubled, so is the maximum bending moment, the length being the same, and as the section modulus is also doubled, the stress is $\frac{2Pl}{2Z}=\frac{Pl}{Z}$, the same as before. At the point $\frac{l}{2}$ the bending moment is $\frac{2Pl}{2}=Pl$, so the stress there is the same as at the point of fixation. An exactly similar line of reasoning is applied to Fig. 4, and so on for any number of plates.

If we accept these statements as true—and the deductions are correctly made from the original assumption of the theory—then we must expect, *a priori*, that in any such spring each and every plate is liable to the same mathematical probability of concomitant failure.

How far is this concept verified by experiment? This is the crucial test to which all mechanical theories must submit; and any failure of a theory to explain the test results in a satisfactory manner is proof of its inadequacy. The experimental verification of the theories of leaf springs was made possible by the endurance testing machine.

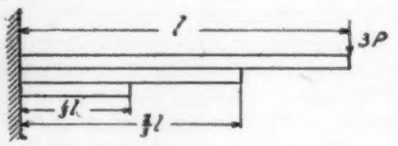


FIG. 4—THEORETICAL THREE-LEAF SPRING

Imagine that we have constructed springs of, say, two, three, four . . . , etc., plates, the master leaf* being the same length in each, and all the leaves having the same cross-section; also that we put these springs into a vibratory testing machine, making each deflect the same distance *d*. What will happen? Since we have conducted exactly such a series of tests, we are able to give a definite reply to this question.

We tested thirty-three springs containing from three to thirteen plates. The results of these tests were most astonishing; in each case, except two, only one leaf in each spring broke, and the broken leaf was always the shortest one! Why? In accordance with the usual theory, all the leaves composing the spring, or at least several of them, should have broken at one and the same time, and the liability of each leaf to fracture should be exactly the same, so that with our considerable number of tests we should have had broken leaves in many positions in the springs. Why this enormous deviation from the results predicted by the theory? We must add another observation of great importance: in no two cases were the number of vibrations required to break the short leaf the same; on the contrary, *the number of vibrations required to break the short leaf decreased as the number of leaves in the spring increased.*

*The longest leaf of a spring, usually having eyes rolled or forged at the ends, is technically called the "master leaf," "main leaf," or "back."

Here we have the results of careful experimental work which seem to indicate the reverse of the theoretical prediction—a theory which has not been questioned since it was first proposed, probably in 1852—sixty-seven years ago. Various explanations were offered to account for these results, and the one which appears to be the most logical and reasonable follows:

Consider, for simplicity, a two-leaf spring such as that shown in Fig. 3, and suppose it to be so constructed that when it is not loaded, or, as we prefer to term it, "free," it is perfectly flat; if arched when free, it does not materially change the reasoning. Now, it is well known to engineers that in the making of a spring each leaf is given a different radius of curvature, as shown roughly by Fig. 5. When the leaves are clamped together by the center bolt, note that they are deflected in opposite directions; the master leaf is deflected upward at the ends, or in the direction opposite to the deflection caused by the external load, while the short leaf is deflected downward, or in the same direction as the deflection caused by the external load. As a consequence, it follows that the short leaf of a spring, no matter what the number of leaves, is thereby already loaded positively. Similarly, the upper leaf is unloaded, or loaded negatively. These initial loading and unloading effects produced by the method of manufacture are present in all commercial leaf springs, though in different degrees.

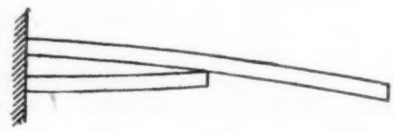


FIG. 5—EXAMPLE OF THE VARYING RADII OF CURVATURE FOR DIFFERENT LEAVES

This method of building leaf springs with initial deflections of the separate laminæ technically called "nip," helps the old theory to explain the breaking of the short leaf first. It is argued that, since the short leaf is already positively stressed by the nipping load, this stress must be added to that produced by the external load, with the result that the short leaf has the greatest stress in it and will therefore be the first to break. This appears to be a fair explanation of the facts brought to view by the destruction tests; but when put to a further test the explanation fails utterly.

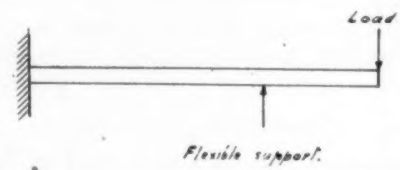


FIG. 6—FUNDAMENTAL PRINCIPLE OF THE AUTHOR'S THEORY

We built approximately the same number and kind of springs as those mentioned previously, but with one important difference; all of these series of springs were built with leaves that were practically "nipless"; in other words, the leaves were curved so that when laid one against the other, without any pressure being applied, the adjacent leaves touched practically everywhere along their entire length, or were, in shop parlance, "dead."

There were, therefore, no initial flexural stresses in these springs.

These nipless series of springs were then placed in the endurance testing machine, with the expectation that several leaves would break at once, and that the fractures would occur in various positions in the springs. What did we obtain? All the nipless springs broke, the short leaf first, just the same as before, although one very noticeable difference was found, namely, that the number of vibrations required to cause fracture was increased. This latter result might, of course, have been anticipated *a priori*, but that the short leaf should continue to break first, and with such repeated insistence, was a surprise.

These facts, constantly shown by the endurance testing machine, and so often confirmed from observation of leaf springs in practice, together with the fact that it has been found necessary in railway as well as in automobile practice to adopt a "constant" in the ordinary formula for the strength of leaf springs which allows for a very nominal working stress, all indicated the necessity for a new analysis, of quite a different character from that of the academic texts.

The new theory has one important element in its favor —it explains exactly everything that has needed an explanation; analyzes fully and completely the minute details of the behavior of each leaf, and the stress at every point in the leaves, and predicts with certainty the endurance of each leaf comprised in a spring. It has explained, easily and consistently, each and every one of the numerous apparent inconsistencies shown by the many tests made, and also the results found "on the road." It has enabled us to build springs having greater endurance than those designed on the basis of the old theory.

New Theory of Leaf Springs

If a series of laminæ having the form of a cantilever, and having equal widths and equal or different thicknesses, be assembled so as to form a leaf spring, and such spring be loaded in the usual manner, it will be found that each leaf touches the one above it only at the point of encastrèment and its extremity. This can readily be shown theoretically, and is obvious in practice, as will be admitted by all who have examined the wear on spring leaves which have been in use for some time—it being understood that leaves with ordinary non-tapered ends are here being considered, and not the special cases to be studied later of the tapered leaf spring.

Any leaf of a spring except the short plate can be considered as a beam, encastrè at one end, loaded at the other, and having a flexible support somewhere between the point of encastrèment and that of application of the load; and this is the fundamental assumption on which our new theory of leaf springs is based. This fundamental principle is illustrated by Fig. 6; its adoption leads to many special and interesting relations, a few of which we will develop.

Let l = the half length of the short leaf for a semi-elliptic spring, or the length for a cantilever spring; l_2 that for the second leaf, etc., so that l_n is the length for the nth leaf.
M = the bending moment at any section
I_n = the moment of inertia of the section of the nth leaf
E = the modulus of direct elasticity of the material
W_n = the reaction, or pressure, between the end of the nth leaf and the $n+1$th leaf
x = the distance of any section under consideration from the point of encastrèment
y = the deflection at any point
δ_n = the deflection at the end of the nth leaf, where W_n acts

The fundamental relation between the curvature (in ordinary cases) and the bending moment is well known to be

$$EI\frac{d^2y}{dx^2}=M \qquad (1)$$

Now, referring to Fig. 7, it will be seen that for the $n+1$th plate, we must have for

$$o \leqq x \leqq l_n$$

$$M = EI_{n+1}\frac{d^2y}{dx^2} = W_{n+1}(l_{n+1}-x) - W_n(l_n-x) \qquad (2)$$

and for

$$l_n \leqq x \leqq l_n+1$$

$$M = EI_{n+1}\frac{d^2y}{d^2x} = W_{n+1}(l_{n+1}-x) \qquad (2a)$$

Integrating, and noting that $EI_{n+1}\frac{dy}{dx}$ from (2) must be equal to $EI_{n+1}\frac{dy}{dx}$ from (2a) when x has the value l_n, we obtain for

$$o \leqq x \leqq l_n$$

$$EI_{n+1}\frac{dy}{dx} = W_{n+1}\left(l_{n+1}-\frac{x^2}{2}\right) - W_n\left(l_nx-\frac{x^2}{2}\right) \qquad (3)$$

and for

$$l_n \leqq x \leqq l_n+1$$

$$EI_{n+1}\frac{dy}{dx} = W_{n+1}\left(l_{n+1}-\frac{x^2}{2}\right) - W_n\frac{l_n^2}{2} \qquad (3)$$

Integrating again, and noting the equality of $EI_{n+1}y$ from (3) and (3a) for $x = l_n$, we obtain for

$$o \leqq x \leqq ln$$

$$EI_{n+1}y = W_{n+1}\left(\frac{l_{n+1}x^2}{2}-\frac{x^3}{6}\right) - W_n\left(\frac{l_nx^2}{2}-\frac{x^3}{6}\right) \qquad (4)$$

and for

$$l_n \leqq x \leqq l_n+1$$

$$EI_{n+1}y = W_{n+1}\left(\frac{l_{n+1}x^2}{2}-\frac{x^3}{6}\right) - W_n\left(\frac{l_n^2x}{2}-\frac{l_n^3}{6}\right) \qquad (4a)$$

Putting $x = l_n$ in (4) and $x = l_{n+1}$ in (4a) we obtain

$$EI_{n+1}\delta_n = W_{n+1}\left(\frac{l_{n+1}l_n^2}{2}-\frac{l_n^3}{6}\right) - W_n\frac{l_n^3}{3} \qquad (5)$$

$$EI_{n+1}\delta_{n+1} = W_{n+1}\frac{l_{n+1}^3}{3} - W_n\left(\frac{l_n^2\,l_{n+1}}{2}-\frac{l_n^3}{6}\right) \qquad (5a)$$

Now putting n for $n+1$ in (5a) there results

$$EI_n\delta_n = W_n\frac{l_n^3}{3} - W_{n-1}\left(\frac{l_{n-1}^2l_n}{2}-\frac{l_{n-1}^3}{6}\right) \qquad (6)$$

and equating the values of δ_n as given by (5) and (6) we obtain, assuming the material of all the leaves to be the same, so that the E's are equal

$$6\,E\delta_n = \frac{W_{n+1}\,(3l_n^2l_{n+1} - l_n^3 - 2W_nl_n^3}{I_{n+1}} =$$

$$\frac{2W_nl_n^3 - W_{n-1}\,(3l_{n-1}^2l_n - l_{n-1}^3)}{I_n} \qquad (7)$$

If the material of the leaves be different, then equation (7) becomes

$$6\delta_n = \frac{W_{n+1}\,(3l^2_n l_{n+1} - l^3_n) - 2W_n l_n^3}{E_{n+1}\,I_{n+1}} = \frac{2W_n l_n^3 - W_{n-1}\,(3l^2_{n-1} l_{n-1} l^3_{n-1})}{E_n I_n} \quad \ldots\ldots (7a)$$

Equation (7a) is of academic interest only, for clearly the material of all the leaves in any spring is always the same; still we wish to indicate the generalization of the theory.

Equation (7) is the most important and fundamental relation between the lengths of the leaves and their corresponding reactions or tip pressures, W. For any particular spring of given dimensions, the reactions may readily be determined in progressive order, starting with the bottom or short leaf.

For the short leaf we have $n = 1$, and $l_o = 0$, so that the relation between the short leaf and the second one is

$$\frac{W_2(3l^2_1 l_2 - l_1^3) - 2W_1 l_1^3}{I_2} = \frac{2W_1 l_1^3}{I_1}$$

and since the l's are usually given, the W's are determined from this relation. Knowing the loads and the lengths, as well as the cross-sections, the stresses are then readily found.

Equation (7) explains the many difficulties and inconsistencies found when endeavoring to force the old theory into agreement with practice; it shows that the strength of a spring is not nearly in direct proportion to the number of plates. It interprets, without any modification, the reason for the failure of the short leaf of any ungraded spring, and of nearly all the graded ones; it shows the reason the endurance at a given deflection decreases with the increase in the number of leaves, and it fully explains many apparent inconsistencies.

Application of Formula

Having now shown the derivation of one of our fundamental formulas, we propose to show its application to the study of particular cases. Not the least of its usefulness will be found in its application to the study of the exact character of the deflection experienced by each leaf of a spring. We shall show further that equation (7) explains fully all the difficulties indicated by the endurance tests mentioned previously, and also many others for which the old theory is unable to offer any rational explanation.

We may mention here that the fundamental reason for failure of the old theory is that it ignores the clamping of the leaves at the center of a semi-elliptic spring, or at the end of a cantilever spring. The old theory assumes that each plate can move vertically and independently at the point of encastrèment, and is quite correct if such a condition holds; this condition, however, never does hold in practice, and the effect of the clamping is sufficient to vitiate the old theory to such an extent as to render it useless from both theoretical and practical points of view.

We will now apply our established equations to see what they may reveal, and as the simplest elementary case let us apply the fundamental equation (7) to a two-leaf spring. We will take the overhangs equal, so as to get an exact comparison with the results of the old theory.

If the plates are of the same cross-section, the moments of inertia are equal; then the fundamental equation becomes

$$W_{n+1}(3l^2_n l_{n+1} - l^3_n) - 2W_n l_n^3 = 2W_n l_n^3 - W_{n-1}(2l^2_{n-1} l_n - l_{n-1}^3)$$

or for the present case, where $n = 1$ and $l_2 = 2l_1$, we have

$$W_2(3l^2_1 l_2 - l_1^3) - 2W_1 l_1^3 = 2W_1 l_1^3 - W_0(3l^2_0 l_1 - l_0^3)$$

and as $l_o = 0$ (has no existence), the last term vanishes, and l_1 cancels, leaving

$$5W_2 - 2W_1 = 2W_1, \text{or } W_2 = 4/5 W_1$$

Here we see that the new theory indicates the reactions to be not at all equal. To put it into words, our theory shows that for a unit load W to act on the point of the short leaf, the load applied to, or the pressure required on the end of the next longer leaf, is only four-fifths of the unit load W_1; or, for unit load placed on the end of the second leaf, in the present case the master leaf, there is produced a reaction or pressure on the end of the short leaf equal to five-fourths times the unit load. This is a very different result from that given by the old theory, and shows that the actual stress in the short leaf is 25 per cent more than is accorded to it by the old theory.

The maximum bending moment on the short leaf is $W_1 l_1$; that on the plate next above it is

$$W_2(l_2 - l_1) = W_2(2l_1 - l_1) = W_2 l_1 = 4/5 W_1 l_1$$

This calculation shows conclusively that the stress in the master leaf of a two-leaf spring, at its maximum value, can be only four-fifths of that in the short leaf. We see thus that the actual stress in the short leaf, according to our theory, is considerably greater, which accounts exactly for the results our experiments persisted in showing and which are so frequently observed in practice; namely, that the short leaf fails first.

Fig. 8 shows graphically the stress at every section of both plates. It will be noticed that the stress in the short lamina increases from zero at the end to a maximum at the point of fixation, while at the same time the stress in the master leaf increases from zero at the end to a maximum, which occurs directly over the end of the short leaf, and then gradually *decreases* toward the point of fixation. The maximum stress in the master leaf is thus only four-fifths of that in the short leaf, and the stress in the master leaf at the point of fixture is only three-fifths of that in the short leaf at the same place. It should be noticed, too, that the distribution of the stress is entirely different for the two leaves—a condition not even suspected heretofore.

Our next point is of still greater interest. It deals with the practical issue of the permissible loads which any spring can safely carry according to this new theory.

Since the short leaf of the two-leaf spring just considered is but one-half of the length of a single-leaf spring, the actual load W_1 which can be placed upon it is evidently equal to $2W_1$, since for the same stress the load varies inversely as the length; hence, to produce double the pressure or reaction on the end of the short leaf we must place on the end of the main leaf four-fifths of the double, or one and three-fifths times the load that we can put on a single-leaf spring. Hence the conclusion from the application of this new theory to a two-leaf spring is that such a two-leaf spring having plates of the same cross-section, and equal steps, is capable of carrying only one and three-fifths times the load of a one-leaf spring. The old theory, then, which states that a two-leaf spring has double the strength of a single-leaf spring is seen to be quite incorrect. It follows, from what we have shown, we are correct in saying that *a two-leaf spring has one and three-fifths times the strength of a single-leaf spring*, and it is in this sense that we shall continue to use the word "strength" as applied to a spring; namely, the ratio of its supporting ca-

pacity (for a given maximum fiber stress in any one leaf) to that of a single leaf with the same maximum fiber stress.

A further application of our theory, quite similar to that just given, shows that a three-leaf spring has not a strength of 2, but one of 2.207; a four-leaf spring has a strength of 2.817, and a ten-leaf spring only 6.503. A twenty-leaf spring has a strength of but 12.689, instead of 20, the reduction in strength being therefore almost 40 per cent. These figures show very definitely the cause of failure from overloading springs designed on the assumption that the strength of such leaf springs varies in direct proportion with the number of leaves; it also shows why, when using the old formulas, it has been found necessary in practice, as mentioned above, to adopt a "coefficient" allowing for an abnormally low apparent stress; the fact being, of course, that the actual stress is very much higher—a condition our theory shows rationally.

In practice it may be observed that for springs having a considerable number of plates there is, in addition to the master leaf, one or even several leaves having the same length as the master leaf; these leaves are called "full-length leaves." Our fundamental equation (7) deals readily with these cases, for any number of full-length leaves are exactly equivalent to a single leaf with a moment of inertia equal to the sum of the moments of such full-length leaves, and the fundamental equation allows for any relation between the moments of inertia of the various leaves.

FIG. 7—APPLICATION OF THE THEORY TO A MULTIPLE-LEAF SPRING

Deflection of Springs

The deflection of a spring has long been a favorite in engineering discussions as a means of verifying the old academic theory of the leaf spring. The practical deflection results having often been found to be in fair agreement with the academic predictions, they have frequently been urged as a confirmation of the theory. Our theory shows that in general the deflection errors of the old theory are much less than the stress errors. The many discrepancies that did occur were explained by a series of carefully phrased objections which have apparently answered the purpose up to the present time. Thus we find that if the calculated deflection was smaller than that found on a test, it was easily explained as due to the friction of the leaves, or to "modulus variation" of the steel, etc. As a matter of fact, carefully conducted tests showed that none of the factors mentioned could explain the large differences often found between the theoretical and the practical deflections. The failure of the old theory is due to its real and inherent defects and not to any imaginary variations in materials, etc. Thus, for example, the old theory supposed that the reactions are equal, which we have shown to be very far from the fact. It necessarily follows, therefore, that there must be analogous errors in the assumption for the deflection formulas.

The old theory shows, for example, that the deflection of a two-leaf spring under a load of $2W$ is the same as that of a single-leaf spring under a load of W. Indeed, the old theory assumes that the deflection is inversely proportional to the number of leaves in the spring, supposing all to be of the same cross-section, or that it is inversely proportional to ΣI at the center of the spring. The old theory does not consider the effect produced by changes in the length of the steps; it does not, in fact, consider the effect of the stepping at all. The mere mention of this is sufficient to show that the old theory is inadequate; a very serious error may easily be committed by ignoring the effect of the stepping on the deflection of a spring, as may be gathered from our fundamental equation (7), which shows that the deflection depends on the length of each and every plate.

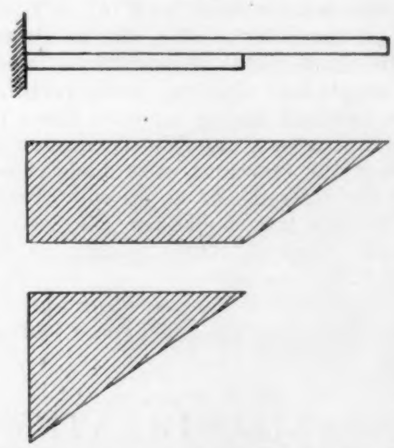


FIG. 8—GRAPHIC DIAGRAM OF STRESSES IN BOTH LEAVES OF A TWO-LEAF SPRING

We have seen that in the ordinary leaf spring with equal steps the reactions are not equal, and the stresses are not equal either. It may therefore occur to the reader that if it were possible to make the reactions

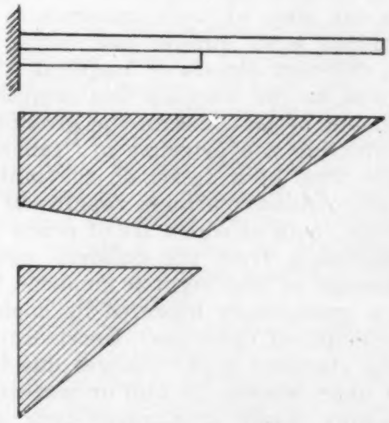


FIG. 9—STRESS DIAGRAM FOR A TWO-LEAF SPRING HAVING EQUAL REACTIONS

equal we might also obtain equal stresses. The attempt would therefore be to question the effect produced by, say, altering the steps, or changing the lengths of the leaves. To make the reactions equal in any spring is easily done by an application of the fundamental equation (7), which now becomes

$$3l_n^2 l_{n+1} - l_n^3 - 2l_n^3 = 2l_n^3 - 3l_{n-1}^2 l_n + l_{n-1}^3$$

or

$$3l_n^2 l_{n+1} = 5l_n^3 - l_{n-1}^2 (3l_n - l_{n-1})$$

for, as the W's are now supposed to be equal, they naturally cancel, and for the present we are considering

only leaves of the same cross-section, so that the I's are also equal. For $n = 1$ this equation evidently becomes

$$3l_1^2 l_2 = 5l_1^3 - l_0^2(3l_1 - l_0)$$

and the last term vanishes as before, since $l_o = 0$, and on dividing by l^2_1, there results

$$3l_2 = 5l_1 \text{ or } l_2 = \tfrac{5}{3}l_1$$

and this is the answer to the question proposed for the two-leaf spring. For further reference we may consider Fig. 9.

The maximum bending moment in the short leaf is Wl_1, and in the master leaf is $W(l_2 - l_1) = 2/3Wl_1$. Now, since l_1 is three-fifths of l_2, the maximum bending moment on the short leaf is three-fifths of what it would be with a single-leaf spring, and, reciprocally, the strength of a two-leaf spring made to these proportions is five-thirds that of a single-leaf spring, or 66 2/3 per cent more, when the reactions were made equal. This is greater than the strength of an equally stepped spring, because the bottom leaf is longer. Indeed, the longer the bottom leaf (after a certain minimum), the greater the strength of the spring.

The general laws which the preceding study enables us to formulate with regard to non-tapered leaf springs are

1 In any leaf spring having leaves of equal cross-section, and with equal steps, the reactions or pressures between the leaves continually decrease from the short leaf toward the master leaf, as do likewise the stresses.

2 In any leaf spring having leaves of equal cross-section, and in which the reactions or pressures between the leaves are equal, the steps or overhangs continually decrease from the short leaf toward the master leaf.

In Fig. 9 we show graphically the result of making the reactions equal and its effect on the stress distribution. The stresses are greatly altered, and are made uniform in the longer leaf for a distance of three-fifths of its length. This graphic analysis shows also why we can carry a greater safe load on this spring than on the spring with equal steps shown in Fig. 7, for we stress the main leaf more uniformly throughout its length, and so place a larger volume of metal at work; at the same time, the short leaf being longer, there is a greater total weight of metal in the spring, and actually the increase of strength is not in proportion to the extra weight of metal, for the increase of strength is only 4.1 per cent, while the increase of weight is 6.7 per cent.

Observe in Fig. 9 that the *stresses are still the greatest in the short leaf*. It is impossible to alter this by merely changing the steps.

STANDARDIZATION IN MERCHANT SHIPBUILDING

FROM the shipbuilders' point of view the advantages to be gained by an efficient system of standardization are many. Owing to the handling of the same class of materials, yard plant may be reduced to a minimum and may be run at its maximum efficiency. Templates for the various parts of the ship's structure may be used indefinitely—sheet iron being substituted for the usual wood of which templates are made. Workmen become specialists in the class of work undertaken, and should therefore produce work quicker and better than when dealing with different classes of material. The system lends itself well to the working and completion of the maximum amount of material on the ground and under cover, thus utilizing the principle of "fabrication" and shortening the time of a vessel on the building berth. These items all combine to achieve rapidity of output and low building cost, both of which are of prime importance.

The disadvantages from the builders' point of view are serious enough at first sight to condemn the project, and they have undoubtedly hindered its adoption in the past. The principal of these are: Uncertainty of repeat orders for any standard type; changes due to evolution of type or to improvements in hull or machinery. Each shipping company wants a standard type of its own. With regard to the first of these disadvantages, very few companies order more than one or at most two ships at a time, and in the case of those companies which make a practice of building new vessels at regular intervals, it is usually found that each vessel is larger than the preceding one or is radically altered in arrangement. Under the second disadvantage may be classed such structural alterations as those involved in the change from the ordinary transverse to the longitudinal system of framing and the structural changes involved in the change from reciprocating steam engines to oil engines for motive power. Under the third disadvantage may be grouped some points which weigh heavily against standardization and arise from consideration of the number and variety of trades that had been developed by shipowners prior to the war. Length of voyage, nature of cargo and number of passengers carried, depth of water and cargo and coaling facilities available at ports of call; all these and many other considerations determine the dimensions and design of merchant vessels of all classes, and where more than one company is engaged in the same trade there is inevitably competition as to which shall own the largest and fastest vessels in the service. This competition leads to constant change in the designs of vessels built by any one company for a particular trade, and tends very directly to prevent any one type becoming standardized.

As regards design of hulls and speed of ships it is obvious that under normal competitive conditions but little approach to standardization is possible, and if, as seems likely, post-war conditions rapidly become of the same competitive character, the prospects for the general adoption of standardization are poor. Propelling machinery of suitable power for each new design of vessel will be required, and here, again, the varieties of power required will preclude the general adoption of any standards. While the prospects of standardization as applied to hulls and propelling machinery are thus very doubtful, there is no question that much could be done to standardize many of the deck fittings, etc., on vessels and much of the auxiliary machinery.

Conditions immediately after the war as regards the urgent necessity for new tonnage will probably be such that owners will be glad to order standard ships if these are reasonably near their requirements to insure the more rapid delivery of such vessels from firms now engaged upon that class of work. The severe internal competition which will subsequently have to be met by British shipping companies would, however, point to the conclusion that the only prospect of real development in standard shipbuilding lies in the elimination of the competition between shipping companies through the formation of combinations controlling, within limits, freights on all the main trade routes.

—*Engineering* (London)

Training in Motor Mechanics for Crippled Soldiers

By DOUGLAS C. MCMURTRIE * *Illustrated with* PHOTOGRAPH

MOTOR mechanics has proved a most popular subject of instruction for crippled soldiers who are being restrained to become self-supporting, self-respecting workers. In fact, the trade is almost too popular, say directors of Canadian schools in which hundreds of disabled soldiers are today being trained in new occupations.

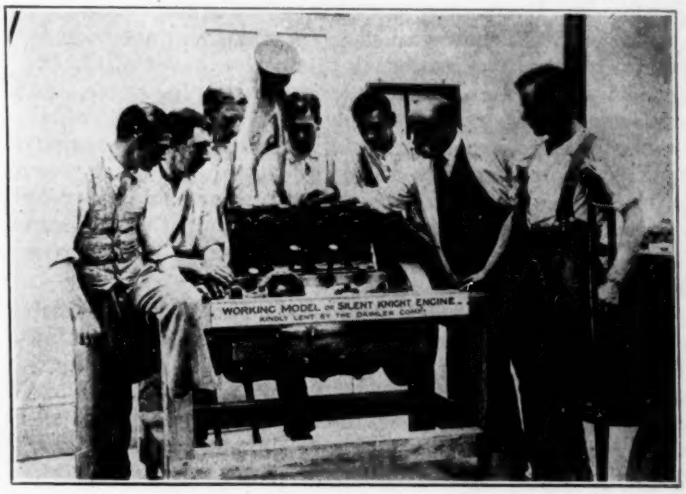
Practically every Canadian soldier who is asked to choose from among the various trades in which classes are operated selects automobile mechanics, but most of the men have to be dissuaded from their intention and turned into other channels of industry. Otherwise, every crippled soldier in the Dominion would be looking for a job as automobile mechanic after he completed his course of training, and the supply would greatly exceed the demand.

Canada, England and India are among the belligerents who offer training courses in motor mechanics to the disabled men of their own forces, and America, in line with the program of reeducation that she has adopted, is teaching her wounded and disabled soldiers automobile repairing. At Fort McHenry, where the United States operates a large reconstruction hospital, this is one of the trades taught to convalescent soldiers. A one-armed or one-legged automobile mechanic will not be a new thing under the sun, once the general public has been committed to the principles of reeducation.

In British Columbia gasoline-engine classes were organized soon after the wounded began returning from overseas. Vancouver, Victoria, Esquimault and Westhaven provide instruction in motor mechanics. In Victoria the Military Hospitals Commission and the Board of Education together operate a fully equipped motor repair shop in which men who wish to become chauffeurs are taught. Men trained in this course conduct a well patronized jitney stand in the town. Those who wish a thorough course preparing them as motor mechanics are sent to Vancouver and later to the new workshops at Esquimault and Westhaven. The Military Hospitals Commission has made an arrangement by which war cripples in Vancouver are taught driving at an excellent automobile school in evening classes, and through the generosity of the owner of this school, any disabled soldier may also attend the day classes free of charge. Several men from other parts of British Columbia are kept in Vancouver on a maintenance allowance while receiving training.

There is a big field for workmen in this trade in Saskatchewan, as through all the west, and thorough courses are given for war cripples by the University of Saskatchewan. Another course is at the Provincial Institute of Technology and Art at Calgary. Disabled men so trained compete successfully with the average sound mechanic. One soldier who lost his right arm is preparing with his chum to have a small flour and grist mill in one of the centers of the Peace River district. His course in automobile mechanics and gas engineering will enable him to take charge of the stationary gas engine powerplant, to run a wood-cutting plant and to mend automobiles as a side line. His chum will attend to the milling proper.

*Director, Red Cross Institute for Crippled and Disabled Men.



A GROUP OF PARTIALLY DISABLED SOLDIERS LEARNING TO BECOME AUTOMOBILE MECHANICS AT AN ENGLISH RECONSTRUCTION HOSPITAL

While the wounds of English soldiers and sailors are healing at Queen Mary's Convalescent Hospitals in Brighton and Roehampton, the men are afforded the opportunity of learning motor mechanics. The workshop at Roehampton is fitted up as a model garage in charge of disabled men who, besides being skilled instructors, understand the special problems of the war cripple. A chassis, working models of engines, three center drilling and turning machines, a drilling machine and benches fitted with the vises and tools needed in repair work are included in the equipment.

Even in far-off India the trade of automobile mechanic has caught the imagination of the natives. It must be a curious sight indeed to see India's disabled sons repairing an automobile or studying the intricacies of mechanics in the shops that are operated at Queen Mary's Technical School in Bombay. These curly-bearded, olive-skinned warriors will not be left on the highroads to beg, after they have served their country, but will be trained for useful trades in which their physical handicaps do not prevent them from competing with able-bodied men.

COOPERATION

COOPERATION has been the dominant note that has made victory possible. Cooperating under General Foch, the United Armies won; the cooperative effort of the Allied navies kept the seas free for commerce; and behind the lines at home the splendid cooperation of industry, of labor and of capital supplied the armies and the navies with all that made that victory possible. It has taken a world-war to bring the lesson home to us. The case has been proved; the demonstration made; it remains for us to make use of knowledge so dearly bought.

—M. L. Requa.

Mexico as a Source of Petroleum and Its Products

By R. De Golyer * (*Non-Member*)

Annual Meeting Paper

MEXICO, with its production of approximately 67,000,000 bbl. in 1918, apparently achieved second place among the petroleum-producing nations of the world. The United States, with a marked production of some 345,000,000 bbl., was secure in first place, but it is certain that revolution-ridden Russia could not have produced enough of its normal 60,000,000 to 70,000,000 bbl. to enable it to retain second place.

This position, now gained by Mexico, will not soon be relinquished. The potential production since 1911, the year in which Mexico became an exporter of petroleum, has been far in excess of the actual production, which in the past few years has been limited by the serious tank steamer shortage resulting from the great war. With the ending of the war, the tankers are being rapidly released and many of them are going into the Mexican trade. Production for the present year is likely to be greatly in excess of that of 1918.

There are two general regions in Mexico from which petroleum has been produced—the highly important Tampico-Tuxpam region and the less explored Tebuantepec-Tabasco region. The Tampico-Tuxpam region, which includes the section of the Gulf coastal plain adjoining the ports of Tampico and Tuxpam, is the region from which practically the entire commercial production of Mexico comes at the present time.

The fields of the Tampico-Tuxpam region are divided generally into two groups—those of the Panuco River valley region and those of the southern or Tuxpam region. The fields of the Panuco River valley region, including the Panuco, Ebano-Chijol, and Topila pools, produce heavy viscous petroleum of 10 to 13 deg. Baumé gravity which are used principally in their crude state as fuel oils. The fields of the Tuxpam zone, including Potrero del Llano, Casiano-Tepetate, Cerro Azul, Los Naranjos, Alamo, and Furbero pools, produce lighter petroleums of 19 to 22 deg. Baumé gravity which are the Mexican petroleums used generally for refining purposes.

Approximately 69 per cent of the petroleum produced in Mexico in 1917, the last year for which detailed statistics are as yet available, was of this grade, and 31 per cent was of the heavier Panuco grade. The proportion of the lighter crude was probably even greater in the production of the past year. Of the 1917 Mexican petroleum production, some 77.6 per cent was exported. Exports for the past year show an even greater percentage and will increase as the Mexican production increases. Of the petroleum remaining in the country during 1917, the equivalent of 5.2 per cent of the total production represents fuel consumed by the Mexican railways and 1.5 per cent represents petroleum consumed principally as fuel in the industry itself. The remaining 15.5 per cent of the total production includes petroleum and products consumed in Mexico, refining losses, increase in storage, if any, etc.

Export Trade with the United States

The United States is the greatest single market for Mexican petroleum. In spite of limited transportation facilities during 1917 because of tanker shortage, the United States took petroleum and products from Mexico equal to 65.9 per cent of its entire production. Other nations took 11.7 per cent. The imports of crude petroleum, distillates, and various refined products from Mexico to the United States during that year were equal to more than 10 per cent of the entire production of the United States in its banner year, the one just past.

The benefits resulting from this condition are reciprocal. The United States profits by getting the petroleum, and Mexico, by the nearness of a great market where, as a result of experience acquired from the utilization of its own immense petroleum supplies, American industries are accustomed to the use of petroleum and its products to an extent not equaled in any other country of the world.

It has been noted that the potential production of Mexico is far in excess of its actual production. It is estimated that the total capacity of wells already completed in Mexico is more than 1,000,000 bbl. per day. In other words, if the petroleum could be taken care of, so that all the wells could be opened at once, the rate of production would be some eight to ten times the actual present rate. This potential production is slightly greater than the present actual production of the United States. The comparison is likely to mislead, however, unless it is remembered that the production of the United States is an actual proved production and can be maintained for some time by drilling up proved areas, whereas to maintain the actual production of Mexico for a year at its potential capacity would undoubtedly require the discovery of new fields.

We have been so impressed by the unprecedented size of some of the Mexican gushers and by their continued production of large quantities of petroleum over long periods of time without any appreciable decline in amount of petroleum produced daily or in field pressures that we have perhaps overestimated the total amount of petroleum to be secured from any single pool. The explanation of the great gushers seems to lie in the very great porosity of the rock in which the petroleum occurs. It collects in a network of caves and channels previously dissolved out of a bed of very thick limestone by the action of water. This condition allows the petroleum to move about very freely while still underground. Furthermore, the petroleum generally lies over water under an artesian head and as a consequence the field pressure is largely hydrostatic rather than gas pressure, which in most oil fields is the expulsive force causing the oil to flow. Effectively, the result of these conditions seems to be that in Mexico there are deposits of petroleum which can be exhausted with a single well, whereas a deposit of the same size under different conditions of occurrence would require hundreds if not thousands of wells to

*Consulting Geologist, New York City.

exhaust it. For comparative purposes it might be noted that there are two wells in Mexico, Potrero del Llano No. 4 and Juan Casiano No. 7, either of which has produced more petroleum than any single field along the Gulf Coast of the United States, while the production of the biggest fields of the Gulf Coast has come from hundreds, if not thousands, of wells, in each instance. The gusher condition in Mexico seems to indicate ease in exploiting rather than such abnormally large pools as have been inferred from the great size of the gushers encountered.

Development Since 1910

Until 1910, both the actual and potential production of Mexico were almost insignificant, in fact, not great enough to supply the domestic trade of Mexico itself. Petroleum was imported from the United States and refined at the Minatitlan plant of the Mexican Eagle Oil Co. and the Tampico plant of the Waters-Pierce Oil Co. Small amounts of petroleum were produced at Furbero and in the Isthmus of Tehuantepec and refined by the Mexican Eagle Oil Co. at Minatitlan. A small amount of very heavy petroleum produced at Ebano by the Mexican Petroleum Co. was being topped in a small field plant. The distillate was sent to the Tampico refinery for further treatment and the residue made into asphalt or used as fuel on the Mexican railways. The highly important Dos Bocas and Casiano fields had been discovered, but Dos Bocas had been lost by fire and the discovery wells in the Casiano field had fallen off in production until there was some doubt whether they would be able to supply enough petroleum to run the pipe line then under construction to Tampico.

During 1910, however, the Potrero and Tanhuijo fields were discovered and Potrero No. 4, which has since produced more than 100,000,000 bbl. of petroleum, was brought in. Juan Casiano No. 7, with a record second only to that of Potrero No. 4, was completed and the discovery well in the Panuco field was brought in.

The potential production of Mexico thus became so great that she had petroleum far in excess of her own requirements—far in excess of the capacity of transportation systems reaching tidewater and thus making the petroleum available for export. This condition has been permanent since that time, so that today the developed production of Mexico is greater than can be carried to tidewater by her rapidly developing pipe-line systems or river-barging equipment. Even if the entire present production could be got to tidewater, it is doubtful whether there are enough ships available to distribute it to the world markets or whether the markets could immediately absorb it. Great fleets of tank steamers to carry Mexican petroleum have been built by the Eagle Transport Co., Ltd. (British), and the Petroleum Transport Co. (American), controlled by the Pearson and Doheny interests respectively, the foremost producers of Mexican petroleum.

The members of this Society are doubtless particularly interested in estimates of the future petroleum supply which can be expected to come from Mexico. The future of supply rather than the past is of greater interest to prospective consumers.

Estimating petroleum reserves is under the best of conditions a somewhat uncertain business. There was the old method of calculating the oil content of a field or property from the thickness and porosity of the oil-bearing rock. The estimate so secured was modified by a safety factor of 20 to 50 per cent to cover petroleum which could not be mined, and on the resultant guess was based the best estimate as to petroleum reserves. The correctness of such a form of estimate depends largely upon a felicitous selection of the safety factor.

We do a little better now perhaps by estimating the probable production of wells to be drilled or the reserve remaining in wells already producing by comparison with the production of average wells in the same or in similar fields. Such a study involves the construction of production curves, the various points on a curve being determined by plotting the amounts produced from a single well or an average well as the ordinates, with the fixed units of time in which produced, arranged consecutively, as the abscissas. Data for the construction of such a graph, to be of any value, must show the changes in the amount of unrestricted production of a given well or average well during various units of time.

We can make estimates of reserves in the Mexican fields by neither method. We have no data as to thickness or porosity of the petroleum-producing formations and consequently cannot use the volumetric method. The bulk of petroleum from Mexico has come from wells of such size that only the production from a restricted flow could be utilized. Production curves constructed on such artificially restricted data as are available under these conditions would be almost valueless. Nevertheless, we can make a rough guess as to the fields already producing in Mexico. It seems fair to assume, on the basis of past performances, that the fields already producing in Mexico indicate what one might call a blocked-out reserve of from a half billion to a billion barrels of crude petroleum

Geologic conditions indicate that other petroleum fields of greater importance than those now known will yet be discovered in Mexico. So far as exploratory drilling is concerned, the petroleum regions have been but scratched. Not more than 1000 wells have been drilled in all of Mexico since the earliest attempt to discover petroleum. Included in this are a great number of wells drilled for exploitation purposes in fields already discovered, and a number of wells drilled in Tabasco, the Isthmus of Tehuantepec region and various outlying regions.

Remarkably few wells are being drilled in Mexico when one considers the amount of petroleum produced. According to official statistics, seventy-nine wells were drilled in 1917, and of them forty-three were productive, with an estimated initial output of 235,027 bbl., and thirty-six were dry holes and abandoned. Bardone of the *Oil and Gas Journal*, estimates that twenty-three wells were completed in the first half of 1918, twelve of them being producers, with an estimated initial production of 350,000 bbl. For comparative purposes, it might be noted that 1117 wells were completed in Kansas and Oklahoma in the single month of July, 1918. These successful wells have been in proved pools with the single exception of Molina No. 2, which was drilled during the latter part of 1917 and which was the discovery well of a new field

The greatest needs of the Mexican petroleum industry at the present time are some relief from the continually increasing taxes, which are apparently designed to be confiscatory, and some degree of safety in the petroleum-producing regions in order that much needed drilling of an exploratory nature may be carried on.

The use of Mexican crude petroleum in internal-combustion engines has not yet passed beyond the experimental stage, but more and more crude petroleum is being refined for its light oil products and this forms an increasingly important addition to the world's supply of engine fuel. The Mexican Eagle Oil Co., Ltd., has re-

fineries at Minatitlan and Tampico and a topping plant at Tuxpam. The Waters-Pierce Oil Co. has refineries at Vera Cruz and Tampico. The Standard Oil Co. of New Jersey has a refinery at Tampico. The Texas Co. has topping plants at Port Lobos and Tampico. The Doheny interests have a topping plant at Tampico and an asphalt plant at Ebano. The Atlantic Refining Co. has a topping plant at Port Lobos.

Only the 19 to 20 deg. Baumé petroleums of the Tuxpam region are refined in quantity in Mexico. All of the refineries and topping plants run it except the Tampico plant of the Texas Co., which tops some Panuco crude, the Ebano asphalt plant which runs Ebano crude, and the Tampico refinery of the Waters-Pierce Oil Co., which runs a very small amount of Topila crude besides much greater amounts of Tepetate-Casiano, Naranjos and Potrero crudes.

Panuco crude is used mostly for fuel purposes. It is so viscous that after the very small light oil fraction has been removed, the residue can be handled only with the greatest difficulty and by specially designed equipment. Panuco crude is imported to the United States and, after being mixed with Gulf Coast crudes, is successfully refined. One American refinery is reported to crack Panuco crude, thus securing 12 to 16 per cent of gasoline or engine fuel.

The greatest possibilities for future extended uses of Mexican petroleum seem to lie either in the further perfection and more widespread development of internal-combustion engines using very heavy oils as fuel, or in an improvement of refining methods by which heavy oils can be more easily converted into lighter oil. It is likely that both methods will be utilized. In the past several years the continued development and widespread use of internal-combustion engines has created such a demand for fuel that it has been supplied only by great efforts on the part of the producer and refiner of petroleum. Fortunately for the petroleum industry, this demand has set the mark and the internal-combustion engine has not waited to be assured of a source of supply for a fixed number of years in advance.

As to the great advantage of the use of Mexican petroleum in internal-combustion engines over its use as fuel for boiler installations there can be no doubt. In this connection, one can hardly do better than quote from a recent paper by Lord Cowdray, head of the Mexican Eagle Oil Co., Ltd., and affiliated organizations:

> It should be stated that Mexican oil, especially refined for use in Diesel engines, is now available for motorships. It is possible that the subject of internal-combustion engines for ships has been discussed with an excess of optimism and led to expectations that have not yet been fully realized, but the most conservative observer cannot fail to be impressed by the solid progress already made in this direction, and the utility of the oil engine for moderate-sized vessels seems to be soundly established. The primary advantage is disclosed in the following figures, giving approximately the comparative consumption by main and auxiliary machinery for various types of marine propulsion.
>
> It will be seen that the oil engine can claim the lowest consumption; the vessel's radius is considerably increased. These are factors which will inevitably insure a great future for motorships, and the provision of fuel supplies on an ample scale will accelerate their progress.

	Lb. of fuel per hp.-hr.
Steam engine, coal-fired	1.60
Steam turbine, coal-fired	1.30
Steam engine, oil-fired	1.00
Steam turbine, oil-fired	0.82
Oil engines	0.50

Statistics covering the production of petroleum by years since the beginning of the industry have recently been made public by the Petroleum Commission of the Mexican Government. They show the past history of Mexican petroleum and indicate prospects for future increases in production better than can be done in any other manner.

Year	Bbl.	Metric tons
1901	10,345	1,544
1902	40,200	6,000
1903	75,375	11,250
1904	125,625	18,750
1905	251,250	37,500
1906	502,500	75,000
1907	1,005,000	150,000
1908	3,932,900	587,000
1909	2,713,500	405,000
1910	3,634,080	542,400
1911	12,552,798	1,873,552
1912	16,558,215	2,471,375
1913	25,696,291	3,835,267
1914	26,235,403	3,915,732
1915	32,910,508	4,912,016
1916	40,545,712	6,059,589
1917	55,292,770	8,264,266
1918 (estimated)	67,000,000	10,000,000
Total	289,082,472	43,166,241

It is confidently hoped that the future production of the Mexican fields will bear out fully the continued increase in amount indicated by the foregoing figures and that it will stop the constantly increasing and extremely critical gap between production and consumption in the United States.

PLANT FUEL SAVING

FOR the efficient execution of the program of industrial conservation the engineers of the country have been working through the professional societies and the operating engineers and firemen, and as a result there were at one time 1500 volunteer engineer specialists and powerplant men organized by states, inspecting powerplants, classifying them according to their operating efficiency, and aiding the work of rapid development. As a direct result of the operation of this plan, it is estimated that the total annual saving throughout the country will be about 25,000,000 tons of coal without reducing the output of the factories. Special printed material giving instructions on the proper use of fuel, has been prepared by the United States Fuel Administration and may be obtained free of cost upon application. This work which was begun as a war measure will be continued, it is expected, for some time after peace is declared to conserve the fuel supply.

The Engineering Division of the Motor Transport Corps*

THE vehicles designed by the Engineering Division have had to be assembled in a large number of plants. The parts making up these vehicles have been made by some hundreds of different manufacturers whose products have had to fit properly. As a result extreme care has been necessary not only in the selection of drafting-room methods but also in the designs themselves, which have necessarily had to be submitted to the closest scrutiny and test before being put into production.

The present article will describe the more important activities of the Design and Records Sections of the Engineering Division. The former, it will be remembered, handles all motor transport design, from the preliminary sketches until the drawings are completed for use in production. The Records Section, in a sense, is the point of contact between the outside world and the Design Section. It supplements the work of the Design Section and in addition prepares and distributes, in accordance with a carefully arranged system, the blueprints, spare parts catalogs and instruction books required in the work of the Engineering Division.

The original design of the Class B truck was undertaken by a number of committees, there being a committee for the engine, another for the axles, a third for transmission, clutch and controls, still another for the general chassis and so on. Each of these committees did its work more or less independently, so that the limits, tolerances and drafting-room practices for which they were responsible varied considerably on the finished drawings. As an instance of the inconsistencies, three or four of the committees used 3/8-in. square keys. In no two cases were the dimensional limits for these keys alike, and consequently a different set of broaches would have been required for every hole and different key stock would have to be purchased for every one of these 3/8-in. keys. Bolt heads, types of thread and limits on reamed holes displayed the same lack of standardization. Further, the analyses of steels and the heat treatments required were not properly standardized or specified in a uniform manner. The design of the Class B truck in general was good but it was not closely knit as regards standards and interchangeability of tools, with the result that the cost of manufacture would have been high.

Function of the Design Section

The function of the Design Section, therefore, as at first organized, was to make the revisions needed to secure uniformity in limits, fits, tolerances, steel and heat treatment specifications in the drawings for the experimental Class B trucks. This function has changed gradually, however, until it is now responsible for the design of all motor transport equipment. As a result, the methods of distributing work in the drafting-room and the type of drawings produced have also been changed.

At first the drafting-room force was divided into squads, one squad being assigned under a chief for each of the major divisions, such as the engine, axle or transmission squads. This type of organization was maintained during the design of the Class B chassis and of the original Class A and AA chassis. In the meantime a great deal of miscellaneous work on bodies, trailers and tank trucks was handled by the body squad. With the expansion of this miscellaneous work it was necessary to change the drafting-room system, and the squads would handle a particular vehicle or type of vehicle, as Class A or Class B trucks, trailers or bodies instead of a particular unit.

Two drafting systems have been developed in the Design Section. In the first, used for chassis and metal units or parts that must be interchangeable, detailed drawings are made for every part of every unit. The complete unit is shown in an assembly drawing which must include every detailed part of the unit. In addition there are made the sub-assembly drawings required to show the construction clearly, as in the case of two welded pieces. The second system, used for bodies and other parts that were built largely of wood, in which the pieces need not be interchangeable, makes use merely of working assembly drawings, these being supplemented by detailed drawings of castings or forgings, parts on which interchangeability can be maintained. The first system cannot be used on body work because of the large number of minor items, such as tacks, thread, turnbuckles, straps and nails, required in the construction.

The Engineering Division has turned out hundreds of designs for experimental work or for production in the past year and a half. Complete working production drawings have been prepared for the Class B (3-ton) chassis, the original Class A (1½-ton) chassis, the original Class AA (¾-ton) chassis, and the Class B, A and AA transport bodies, three sizes of two-wheel cargo trailer, five different tank truck bodies and bodies especially designed for portable machine shops, wrecker trucks and horse transport. Just before the signing of the armistice the Section issued production drawings for the Class AT (four-wheel steer) and TT (two-wheel steer) four-wheel drive trucks. Very recently drawings for the Class A White and Class AA G. M. C. trucks as standardized for military service have been prepared. These vehicles were to be built in a number of different factories, so it was necessary to have drawings and parts lists in the standard form used by the Engineering Division and also as a permanent record of their construction.

The extent of the work done with trailers is shown by the fact that at present there are thirty-eight trailers on the list of vehicles standardized by the Motor Vehicle Board. Trailers of from 1 to 5 tons capacity have been completely designed by the Section, these including the two-wheel, four-wheel and semi-trailer types. The drawings of several trailers developed by the Signal Corps have been revised. Two-wheel trailers have been favored in the smaller sizes. This type has disadvantages but it will follow the truck and operate safely at high speeds when the four-wheel trailer will not track properly. Two models of motor sleighs for use in Alaska and under similar conditions have been developed through the layout stage. One of these was intended for trail breaking and the other for transporting freight after the trail had been broken.

*Second of a series of articles relating to the organization and activities of this branch of the Army. The first article dealing with the subject in general appeared in the January issue of THE JOURNAL, and the next article will relate to the Experimental and Testing Section and to the Technical Service Branch. The source of the articles is the Engineering Division of the Motor Transport Corps.

The design of the Class B truck is an example of the policy followed by the Division. Sound commercial practice has been adhered to in the belief that apparatus for war service should contain no untried or doubtful experimental features. The principal task has been to adapt commercial designs to the severity and peculiarity of military service. In the Class B chassis, for instance, all the parts are of exceedingly liberal size. The power and low gear reduction are greater in Class B than in commercial trucks. The ground clearance is higher, particularly in the center of the vehicle to prevent any of the parts dragging in going over bad ridges. All parts are mounted with extreme care to avoid strains from the distortion of the frame arising from inequality of the road. A system of automatic spring-bolt lubrication has been applied successfully to the Class B truck, the lubricant carried being sufficient for several hundred miles' operation. Many of the features of design of the Class B chassis were incorporated also in the original Class A and AA vehicles. The two smaller vehicles in general represented good commercial practice and contained no unusual features.

In the design of special bodies noteworthy results have been accomplished with tank trucks; two sizes of gasoline tank trucks of 1000 and 180 gal. capacity have been standardized as developed by the Design Section. The members of the Engineering Division, after conference with interested manufacturers, have designed a new manhole which has been standardized for use on army tank trucks. An emergency valve which is operated from the outside of the tanks has also been developed. This is somewhat slower in operation than the commercial product available but has been found to give better general results. One of the most difficult parts of the tank truck design was the rear bumper. This had to be applied so as to clear the tank faucets and at the same time protect the rear end of the tank. The problem was solved by attaching the bumper to the ends of the body sills in such a way that it rests on the chassis frame which therefore absorbs all the shocks.

Designing a New Carbureter

When the Class B truck was being designed, a number of different types of commercial carbureters were tested. It was soon realized, however, that the military service would require a stronger and more economical device than was available. The commercial products on the market were usually adaptations of passenger-car carbureters. The Engineering Division started work, therefore, on the design of a device to be used primarily with motor-truck engines. In a very short time a design was completed, samples built and tested and the changes made to secure a working product, the result finally being the U. S. A. or composite carbureter. The term "composite" is used because it incorporates a large number of unpatented principles of construction found in other carbureters. The tests of different experimental engines fitted with the U. S. A. carbureter have shown that it results in higher mean effective pressures, greater torque and horsepower and lower fuel consumption than any of the commercial models previously tested. Favorable results were also obtained in extended road tests; hence the carbureter has been adopted as standard and was used on the second series Class B trucks.

The U. S. A. carbureter is of plain tube design, a by-pass being provided at the throttle for idling. A metered supply of fuel is admitted to the main nozzles from the float-chamber through one gasoline passage. This nozzle is of the air-bleed type arranged to admit both air and fuel to its mixing chamber. No movable parts are used, the fuel mixture being effected by moving air streams rather than by mechanical agents. A single jet and a single air passage are features of the design, as are also its general strength and durability. All the important parts, such as the throttle and choker shafts, are larger than those usually found in commercial practice, the throttle-shaft turning in bronze bushings and the float reinforced at both top and bottom. While the carbureter was first applied to the Class B engine, it was designed so that it could be readily used with smaller engines. The Class B type contains only fifty-one parts, of which forty-one are used interchangeably in the smaller sizes that have been successfully developed for Class A and AA engines.

In the early part of 1918 a special design of magneto was developed by the Engineering Division for the Class B truck. Samples of this magneto gave satisfactory results on test but it was never put into production because of the time it was thought would be required to overcome the usual initial manufacturing difficulties. The work on this special design was carried on after arrangements had been made to secure an interchangeable product for the first series Class B trucks. Mounting dimensions and those involving attachment to other parts were standardized. These are a compromise based on manufacturing requirements of dimensions used in good commercial practice. The product of the different makers of the Class B magneto has conformed to these standard dimensions, but each maker has been allowed to follow his own practice as to the individual details of the instrument.

The hood-latches used in commercial practice have not been found heavy enough for military truck work. In travel over uneven roads the truck frames weave so that the hood-latch spring must be capable of expanding some ¾ in. after reaching its assembled position. Another requirement is a swivel base so that if the hood shifts forward or backward slightly the pivot-pin of the latch will not bind. In the new latch developed by the Design Section the spring has a movement of 1½ in., a swivel base permits weaving of the hood, and the construction throughout is heavy. The latch stands at an angle from the vertical such that it not only pulls the hood down but also pulls it inward against the ledge.

The Design Section has developed a somewhat unusual front bumper for the Class B and Class A trucks. In this a heavy wood beam is reinforced by structural channels. The bumper in turn is attached to the truck through heavy springs on each side of the frame. The springs absorb any blows received on the bumper, thus dissipati g the shock and preventing damage to the chassis. Before the springs are closed solidly a blow of about four tons can be given the bumper. This design has been found especially useful for military vehicles operating in convoy formation.

Methods of Specifying Steel

It is the common practice of many commercial manufacturing organizations to leave certain questions relating to construction of their product open to the discretion of the shop. The bore for a bearing may be given to the nearest dimension and a note added that it is to be fitted to such and such a bearing. In other cases the drawings carry no limits, the shop being depended upon to see that parts fit properly and that the correct limits are maintained. In the specifying of steels, both as to

TABLE 1 DIRECTIONS FOR HEAT TREATMENT OF STEEL—STANDARDIZED BY ENGINEERING DIVISION, MOTOR TRANSPORT CORPS

Heat Treatment	Operations 1	2	3	4	5	6	7	8
A	Carbonize 1600-1700 deg. fahr. to give depth of case required	Quench from pots in water	Draw 15 min., 350-400 deg., fahr.	Cool in air				
A-1	Carbonize 1600-1700 deg. fahr. to give depth of case required	Quench from pots in oil						
B	Carbonize 1600-1700 deg. fahr. to give depth of case required	Cool slowly in carbonizing material	Reheat 10-20 min., 1425-1475 deg. fahr.	Quench in water	Draw 20 min., 350-400 deg., fahr.			
B-1	Carbonize 1600-1700 deg. fahr. to give depth of case required	Cool slowly in carbonizing material	Reheat 10-20 min., 1425-1500 deg. fahr.	Quench in oil	Draw 20 min., 350-400 deg., fahr.			
C	Carbonize 1600-1700 deg. fahr. to give depth of case required	Cool slowly in carbonizing material	Reheat 15-30 min., 1550-1575 deg. fahr.	Quench in oil	Reheat 15 min., 1450-1475 deg. fahr.	Quench in water	Draw 20 min., 350-400 deg. fahr.	Cool in air
C-1	Carbonize 1600-1700 deg. fahr. to give depth of case required	Cool slowly in carbonizing material	Reheat 15-30 min., 1550-1575 deg. fahr.	Quench in water	Reheat 15 min., 1450-1475 deg. fahr.	Quench in water	Draw at 350-400 deg. fahr.	Cool in air
D	Heat in cyanide 10-20 min., 1550 deg. fahr.	Quench in water						
E	Heat 10-20 min., 1450-1550 deg. fahr.	Quench in oil	Reheat 10-20 min. at temperature to give hardness required.	Cool in air				
E-1	Heat 10-20 min., 1500-1550 deg. fahr.	Quench in oil	Reheat 10-20 min. at temperature to give hardness required.	Cool in air				
E-2	Heat 10-20 min., 1550-1600 deg. fahr.	Quench in oil	Reheat 10-20 min. at temperature to give hardness required.	Cool in air				
F	Heat 10-20 min., 1450-1500 deg. fahr.	Quench in water	Reheat 10-20 min. at temperature to give hardness required.	Cool in air				
F-1	Heat 10-20 min., 1500-1550 deg. fahr.	Quench in water	Reheat 10-20 min. at temperature to give hardness required.	Cool in air				
F-2	Heat 10-20 min., 1550-1600 deg. fahr.	Quench in water	Reheat 10-20 min. at temperature to give hardness required.	Cool in air				
G	Heat 10-20 min., 1500-1550 deg. fahr.	Quench in oil	Reheat 10-20 min., 1450-1500 deg. fahr.	Quench in water	Reheat at temperature to give hardness required	Cool in air		
G-1	Heat 10 min., 1550-1600 deg. fahr.	Quench in oil	Reheat 1450-1500 deg.	Quench in water	Reheat 10-20 min. at temperature to give hardness required.	Cool in air		
G-2	Heat 10-20 min., 1500-1550 deg. fahr.	Quench in oil	Reheat 10-20 min., 1400-1450 deg. fahr.	Quench in water	Reheat 10-20 min. at temperature to give hardness required.	Cool in air		
H	Heat 10-20 min., 1500-1550 deg. fahr.	Quench in water	Reheat 10-20 min., 1400-1450 deg. fahr.	Quench in water	Reheat 10-20 min. at temperature to give hardness required.	Cool in air		
J	Heat 10-30 min., 1500-1550 deg. fahr.	Cool slowly	Reheat 10-20 min., 1450-1500 deg. fahr.	Quench in oil	Reheat 10-20 min. at temperature to give hardness required.	Cool in air		
K	After shaping or coiling: Heat 5-15 min., 1425-1475 deg. fahr.	Quench in oil	Reheat 5-15 min. at temperature to give temper required	Cool slowly in air				
L	Heat 30 min. at 1475-1525 deg. fahr. to insure thorough heating	Cool slowly						

composition and heat treatment, ordinary drafting-room practice is equally indefinite. For instance, a draftsman will often specify that a part shall be made from "machine steel," "screw stock," "cold-rolled steel," or "tool steel," all of which terms are indefinite. Little or no trouble results in commercial practice from the use of such a method, because the engineering department usually knows from previous experience the quality of the metals

and the results that can be expected from the use of any given term. Likewise details of heat treatment are often omitted from drawings, which state simply that the part is to be heat-treated, annealed or case-hardened, the furnace foreman being depended upon to see that the proper treatment is given.

When parts are made in some hundreds of shops all over the country, a very different problem has to be faced. Exact and detailed information regarding a material must be given on all drawings. The first step taken by the Engineering Division in supplying such information was to make use of S. A. E. steel specifications and heat treatments which are, of course, well known to most steel makers or to others who supply raw materials needed for steel parts. In some special cases the chemical analysis of the steel has been given but as a rule it has been the practice of the Design Section to use S. A. E. steel specifications and heat treatments. Material would be indicated by "S. A. E. Steel No. 1035, S. A. E. Heat Treatment H, draw to give 32 to 40 scleroscope." The drawing temperature would then be selected to give the desired hardness. The depth of case required would be given for case-hardened parts. It may be said that the physical properties required have been checked by determinations of elastic limit and also of hardness, using either Brinell or scleroscope instruments for the latter. If the elastic limit and hardness are satisfactory, it is considered that the other physical properties will be also.

While the use of S. A. E. specifications has greatly assisted the manufacturers supplying military truck parts, there has been difficulty, especially when up-to-date heat-treatment facilities were not available, in obtaining steels with uniformly high physical properties. In the military service it is especially important, of course, that similar parts give the same high-grade performance.

To take care of this situation, a large number of tests have been conducted for the Engineering Division, and a careful study made of the performance of different parts with given heat treatments. With this knowledge as a foundation, the S. A. E. specifications have been amplified and complete directions for heat treatment have been prepared. These have been applied in the production of parts and have resulted in complete satisfaction. The steels used must, of course, have the proper physical properties when in the unannealed condition to obtain the correct results.

TABLE 3—CHEMICAL ANALYSIS AND PHYSICAL PROPERTIES OF CARBON STEEL

Carbon	Manganese	Silicon	Phosphorus	Sulphur, Max.	Annealed				Heat-Treated					
					Tensile Strength	Elastic Limit	Red. Area, per Cent	Elon., 2 In., per Cent	Tensile Strength	Elastic Limit	Red. Area, per Cent	Elon., 2 In., per Cent	Brinell Reading	Scleroscope Reading
0.05	0.65	0.16	0.045	0.045	42,800	29,500	63	37	44,800	31,400	67	23	165	25
0.06	0.65	0.16	0.045	0.045	43,500	30,000	62	36	45,800	32,100	66	22	178	27
0.07	0.65	0.16	0.045	0.045	44,125	30,400	62	36	47,600	33,600	66	22	191	29
0.08	0.65	0.16	0.045	0.045	44,700	30,750	61	35	49,300	34,800	65	22	204	31
0.09	0.65	0.16	0.045	0.045	45,300	31,200	61	35	51,200	35,400	65	21	218	33
0.10	0.65	0.16	0.045	0.045	46,000	31,600	60	35	53,000	36,600	64	21	230	35
0.11	0.65	0.16	0.045	0.045	46,600	32,100	59	34	54,800	37,200	63	21	237	36
0.12	0.65	0.16	0.045	0.045	47,300	32,600	59	34	56,600	39,800	62	20	250	38
0.13	0.65	0.16	0.045	0.045	47,800	33,000	58	33	58,300	40,800	62	20	257	39
0.14	0.65	0.16	0.045	0.045	48,600	33,500	57	33	60,200	42,000	61	20	270	41
0.15	0.65	0.16	0.045	0.045	49,200	33,900	57	32	61,900	43,400	60	20	277	42
0.16	0.65	0.16	0.045	0.045	49,900	34,300	56	32	63,800	44,600	59	19	290	44
0.17	0.65	0.16	0.045	0.045	50,500	34,600	55	32	65,600	46,000	58	19	296	45
0.18	0.65	0.16	0.045	0.045	51,200	35,100	55	31	67,400	47,200	57	19	310	47
0.19	0.65	0.16	0.045	0.045	51,800	35,700	54	31	69,200	48,500	55	18	323	49
0.20	0.65	0.16	0.045	0.045	52,400	36,000	53	31	71,100	49,800	54	18	336	51
0.21	0.65	0.16	0.045	0.045	53,000	36,600	52	30	72,800	51,000	52	18	345	52
0.22	0.65	0.16	0.045	0.045	53,700	37,000	51	30	74,600	52,300	51	17	345	52
0.23	0.65	0.16	0.045	0.045	54,300	37,300	50	30	76,300	53,500	49	17	356	54
0.24	0.65	0.16	0.045	0.045	54,900	37,800	50	29	78,200	55,000	48	17	369	56
0.25	0.65	0.16	0.045	0.045	55,500	38,200	49	29	80,000	56,100	47	17	373	58
0.26	0.65	0.16	0.045	0.045	56,200	38,800	48	28	81,800	58,000	46	16	396	60
0.27	0.65	0.16	0.045	0.045	56,800	39,100	47	28	83,600	59,500	44	16	416	63
0.28	0.65	0.16	0.045	0.045	57,400	39,500	46	27	85,300	60,500	43	16	429	65
0.29	0.65	0.16	0.045	0.045	58,000	40,000	46	27	87,200	61,800	42	15	441	67
0.30	0.65	0.16	0.045	0.045	58,700	40,500	45	26	89,000	62,500	42	15	461	70
0.31	0.65	0.16	0.045	0.045	59,300	40,800	45	26	90,800	63,800	41	15	475	72
0.32	0.65	0.16	0.045	0.045	59,900	41,300	44	26	92,600	65,600	40	15	495	75
0.33	0.65	0.16	0.045	0.045	60,500	41,700	44	25	94,400	67,200	39	14	514	78
0.34	0.65	0.16	0.045	0.045	61,200	42,000	43	25	96,200	68,500	38	14	526	80
0.35	0.65	0.16	0.045	0.045	61,800	42,600	42	24	98,000	69,600	38	14	547	83
0.36	0.65	0.16	0.045	0.045	62,500	43,200	42	24	99,700	70,500	37	14	554	84
0.37	0.65	0.16	0.045	0.045	63,100	43,500	41	23	101,600	72,000	36	13	567	86
0.38	0.65	0.16	0.045	0.045	63,800	43,900	41	23	103,400	73,500	36	13	573	87
0.39	0.65	0.16	0.045	0.045	64,400	44,400	40	23	104,200	74,200	35	13	587	89
0.40	0.65	0.16	0.045	0.045	65,000	44,800	40	22	105,000	75,000	35	12	593	90
0.41	0.65	0.16	0.045	0.045	65,600	45,200	40	22	108,800	77,000	34	12	600	91
0.42	0.65	0.16	0.045	0.045	66,300	45,800	39	21	111,600	78,500	34	12	607	92
0.43	0.65	0.16	0.045	0.045	66,900	46,100	39	21	112,400	80,000	33	11	620	94
0.44	0.65	0.16	0.045	0.045	67,500	46,500	38	21	114,200	81,000	33	11	627	95
0.45	0.65	0.16	0.045	0.045	68,000	46,800	38	20	116,000	82,500	33	11	627	95
0.46	0.65	0.16	0.045	0.045	68,800	47,300	38	20	117,800	83,500	32	10	634	96
0.47	0.65	0.16	0.045	0.045	69,500	47,800	37	19	119,600	85,000	32	10	634	96
0.48	0.65	0.16	0.045	0.045	70,000	48,200	37	19	121,400	86,500	31	9	640	97
0.49	0.65	0.16	0.045	0.045	70,600	48,600	36	18	123,200	88,000	31	9	645	98
0.50	0.65	0.16	0.045	0.045	71,000	49,000	36	18	125,000	90,000	30	9	645	99

The data in Table 1 form a summary of all the investigations on steel conducted by the Engineering Division. As previously stated, the various heat treatments have been developed from those given in the S. A. E. specifications. M. T. C. heat treatments A or B are simply S. A. E. heat treatments A or B in an improved form, the temperature range perhaps being given more closely or a definite time factor being introduced. In some cases more than one modification has been found necessary; for example, M. T. C. heat treatments E, E-1 and E-2 as given in Table 1 are all developments of S. A. E. heat treatment E. The different modifications of a given S. A. E. heat treatment are used at the dis-

TABLE 2—CHART OF STEEL SPECIFICATIONS, CHEMICAL ANALYSIS AND PHYSICAL PROPERTIES

Class	S. A. E. Spec. No.	Chemical Analysis: Carbon Min.	Carbon Max.	Carbon Desired	Manganese Min.	Manganese Max.	Manganese Desired	Phosphorus	Sulphur, Max.	Nickel Min.	Nickel Max.	Nickel Desired	Chromium Min.	Chromium Max.	Chromium Desired	Vanadium Min.	Vanadium Desired	Silicon Min.	Silicon Max.	Silicon Desired	Physical Properties: Annealed Tensile Strength	Annealed Elastic Limit	Annealed Red. Area, per Cent	Annealed Elon., 2 In., per Cent	M. T. C. Heat Treatments	Heat-Treated Tensile Strength	Heat-Treated Elastic Limit	Heat-Treated Red. Area, per Cent	Heat-Treated Elon., 2 In., per Cent	Brinell Reading	Scleroscope Reading
Carbon steels	1010	0.05	0.15	0.10	0.30	0.60	0.45	0.045	0.050	...	...	...	...	...	...	...	...	...	...	...	46,000	34,500	72	37	A	60,000	39,000	62	30	..	24
Carbon steels	1020	0.15	0.25	0.20	0.30	0.60	0.45	0.045	0.050	...	...	...	...	...	...	...	...	...	...	...	54,400	39,500	68	32	B or H	79,500	49,500	59	20	176	35
Carbon steels	1025	0.20	0.30	0.25	0.50	0.80	0.65	0.045	0.050	...	...	...	...	...	...	...	...	...	...	...	59,470	45,000	66	30	H or D	89,000	59,000	55	17.5	210	37
Carbon steels	1035	0.30	0.40	0.35	0.50	0.80	0.65	0.045	0.050	...	...	...	...	...	...	...	...	...	...	...	64,000	48,000	62	29	H, D or E	104,000	74,000	43	15.5	255	42
Carbon steels	1045	0.40	0.50	0.45	0.50	0.80	0.65	0.045	0.050	...	...	...	...	...	...	...	...	...	...	...	71,300	57,500	54	23	H, D or E	123,000	88,000	36	13.5	290	45
Carbon steels	1095	0.90-	1.05	0.95	0.25	0.50	0.35	0.040	0.050	...	...	...	...	...	...	...	...	...	...	...	79,000	59,500	51	21	F	175,000	120,000	18	6	551	75
Nickel steels	2315	0.10	0.20	0.15	0.50	0.80	0.65	0.040	0.045	3.25-	3.75	3.50	...	...	...	...	...	...	...	...	54,000	42,500	60	32	H or K	107,500	75,000	55	18	321	43
Nickel steels	2320	0.15	0.25	0.20	0.50	0.80	0.65	0.040	0.045	3.25-	3.75	3.50	...	...	...	...	...	...	...	...	57,000	46,000	60	30	H or K	133,000	98,000	54	16.5	339	45
Nickel steels	2330	0.25	0.35	0.30	0.50	0.80	0.65	0.040	0.045	3.25-	3.75	3.50	...	...	...	...	...	...	...	...	63,000	50,000	56	28	H or K	177,000	124,000	53	15.5	342	46
Nickel steels	2335	0.30	0.40	0.35	0.50	0.80	0.65	0.040	0.045	3.25-	3.75	3.50	...	...	...	...	...	...	...	...	68,000	55,000	53	24	H or K	186,000	151,000	51	15	465	62
Nickel steels	2340	0.35	0.45	0.40	0.50	0.80	0.65	0.040	0.045	3.25-	3.75	3.50	...	...	...	...	...	...	...	...	73,000	58,000	50	23	H or K	195,000	179,000	49	13	525	70
Nickel steels	2345	0.40	0.50	0.45	0.50	0.80	0.65	0.040	0.045	3.25-	3.75	3.50	...	...	...	...	...	...	...	...	78,000	62,500	48	21	H or K	212,000	193,000	45	12	570	76
Nickel chromium steels	3120	0.15	0.25	0.20	0.50	0.80	0.65	0.040	0.045	1.00-	1.50	1.25	0.45	0.75	0.60	...	...	...	...	...	62,000	49,000	53	23	H or D	116,000	85,000	48	23	270	36
Nickel chromium steels	3125	0.20	0.30	0.25	0.50	0.80	0.65	0.040	0.045	1.00-	1.50	1.25	0.45	0.75	0.60	...	...	...	...	...	65,500	52,000	50	22	H, D or E	129,000	94,000	47	21	291	39
Nickel chromium steels	3130	0.25	0.35	0.30	0.50	0.80	0.65	0.040	0.045	1.00-	1.50	1.25	0.45	0.75	0.60	...	...	...	...	...	68,500	56,500	48	21	H, D or E	132,000	112,000	45	20	316	42
Nickel chromium steels	3135	0.30	0.40	0.35	0.50	0.80	0.65	0.040	0.045	1.00-	1.50	1.25	0.45	0.75	0.60	...	...	...	...	...	71,300	57,000	46	20	H, D or E	172,000	125,000	43	18	330	44
Nickel chromium steels	3140	0.35	0.45	0.40	0.50	0.80	0.65	0.040	0.045	1.00-	1.50	1.25	0.45	0.75	0.60	...	...	...	...	...	75,000	60,000	45	20	H, D or E	179,000	130,000	41	18	345	46
Nickel chromium steels	3220	0.15	0.25	0.20	0.30	0.60	0.45	0.040	0.040	1.50-	2.00	1.75	0.90-	1.25	1.10	...	...	...	...	...	69,000	55,000	50	21	G, H or D	151,000	110,000	48	23	375	50
Nickel chromium steels	3230	0.25	0.35	0.30	0.30	0.60	0.45	0.040	0.040	1.50-	2.00	1.75	0.90-	1.25	1.10	...	...	...	...	...	72,000	57,500	48	21	H or D	103,000	150,000	46	21	405	54
Nickel chromium steels	3240	0.35	0.45	0.40	0.30	0.60	0.45	0.040	0.040	1.50-	2.00	1.75	0.90-	1.25	1.10	...	...	...	...	...	75,000	59,000	46	20	H or D	224,000	170,000	40	18	435	58
Nickel chromium steels	3250	0.45	0.55	0.50	0.30	0.60	0.45	0.040	0.040	1.50-	2.00	1.75	0.90-	1.25	1.10	...	...	...	...	...	78,000	62,000	42	19	M or Q	260,000	190,000	32	16	480	64
Nickel chromium steels	*3315	0.10	0.20	0.15	0.45	0.75	0.60	0.040	0.040	2.75-	3.25	3.00	0.60	0.95	0.80	...	...	...	...	...	68,000	52,000	49	23	M	130,000	95,000	46	22	300	46
Nickel chromium steels	*3335	0.30	0.40	0.35	0.45	0.75	0.60	0.040	0.040	2.75-	3.25	3.00	0.60	0.95	0.80	...	...	...	...	...	75,000	57,000	47	21	P or R	157,000	115,000	43	18	389	52
Nickel chromium steels	*3350	0.45	0.55	0.50	0.45	0.75	0.60	0.040	0.040	2.75-	3.25	3.00	0.60	0.95	0.80	...	...	...	...	...	81,000	62,000	44	19	R	191,000	140,000	40	14	450	60
Nickel chromium steels	3320	0.15	0.25	0.20	0.30	0.60	0.45	0.040	0.040	3.25-	3.75	3.50	1.25-	1.75	1.50	...	...	...	...	...	59,500	46,000	50	21	L	150,000	110,000	48	23	375	50
Nickel chromium steels	3330	0.25	0.35	0.30	0.30	0.60	0.45	0.040	0.040	3.25-	3.75	3.50	1.25-	1.75	1.50	...	...	...	...	...	67,000	51,000	47	20	P or R	191,000	140,000	46	21	450	60
Nickel chromium steels	3340	0.35	0.45	0.40	0.30	0.60	0.45	0.040	0.040	3.25-	3.75	3.50	1.25-	1.75	1.50	...	...	...	...	...	74,000	56,000	45	18	P or R	232,000	170,000	42	18	479	64
Chromium steels	5120	0.15	0.25	0.20	...	...	...	0.040	0.045	...	...	...	0.65	0.85	0.75	...	...	...	...	...	55,000	42,500	59	28	B	130,000	98,000	54	16	405	54
Chromium steels	5140	0.35	0.45	0.40	...	...	...	0.040	0.045	...	...	...	0.65	0.85	0.75	...	...	...	...	...	63,000	49,000	42	22	H or D	169,000	125,000	40	10	432	59
Chromium steels	5165	0.60	0.70	0.65	...	...	...	0.040	0.045	...	...	...	0.65	0.85	0.75	...	...	...	...	...	71,000	55,000	38	20	H or D	190,000	160,000	32	8	465	62
Chromium steels	5195	0.90-	1.05	0.95	0.20	0.45	0.35	0.030	0.030	...	...	...	0.90-	1.10	1.00	...	...	...	...	...	78,000	59,000	34	18	M, P or R	240,000	180,000	28	8	479	64
Chromium steels	51120	1.10-	1.30	1.20	0.20	0.45	0.35	0.030	0.030	...	...	...	0.90-	1.10	1.00	...	...	...	...	...	82,000	62,000	31	16	M, P or R	275,000	205,000	26	7	500	66
Chromium steels	5295	0.90-	1.05	0.95	0.20	0.45	0.35	0.030	0.030	...	...	...	1.10-	1.30	1.20	...	...	...	...	...	79,000	60,000	28	15	M, P or R	250,000	190,000	23	6	510	68
Chromium steels	52120	1.10-	1.30	1.20	0.20	0.45	0.35	0.030	0.030	...	...	...	1.10-	1.30	1.20	...	...	...	...	...	82,000	62,000	24	13	M, P or R	253,000	200,000	25	7	524	70
Chromium vanadium steels	6120	0.15	0.25	0.20	0.50	0.80	0.65	0.040	0.040	...	...	...	0.80-	1.10	0.95	0.15	0.18	...	...	...	81,000	60,400	56	28	T	116,000	85,000	48	20	390	52
Chromium vanadium steels	6125	0.20	0.30	0.25	0.50	0.80	0.65	0.040	0.040	...	...	...	0.80-	1.10	0.95	0.15	0.18	...	...	...	83,700	61,000	54	26	T	144,000	105,000	46	19	405	54
Chromium vanadium steels	6130	0.25	0.35	0.30	0.50	0.80	0.65	0.040	0.040	...	...	...	0.80-	1.10	0.95	0.15	0.18	...	...	...	84,500	61,500	51	23	T	163,000	120,000	43	16	432	59
Chromium vanadium steels	6135	0.30	0.40	0.35	0.50	0.80	0.65	0.040	0.040	...	...	...	0.80-	1.10	0.95	0.15	0.18	...	...	...	85,000	62,000	48	22	T	185,000	135,000	41	15	450	60
Chromium vanadium steels	6140	0.35	0.45	0.40	0.50	0.80	0.65	0.040	0.040	...	...	...	0.80-	1.10	0.95	0.15	0.18	...	...	...	86,000	62,600	46	21	T	210,000	155,000	39	13	465	62
Chromium vanadium steels	6145	0.40	0.50	0.45	0.50	0.80	0.65	0.040	0.040	...	...	...	0.80-	1.10	0.95	0.15	0.18	...	...	...	86,700	63,200	44	20	U	242,000	180,000	37	12	479	64
Chromium vanadium steels	6150	0.45	0.55	0.50	0.50	0.80	0.65	0.040	0.040	...	...	...	0.80-	1.10	0.95	0.15	0.18	...	...	...	87,200	63,600	39	18	U	288,000	210,000	34	10	510	68
Chromium vanadium steels	6195	0.90-	1.05	0.95	0.20	0.45	0.35	0.030	0.030	...	...	...	0.80-	1.10	0.95	0.15	0.18	...	...	...	90,000	68,200	38	16	U	330,000	250,000	24	8	562	75
Silico manganese steels	9250	0.45	0.55	0.50	0.60	0.80	0.70	0.045	0.045	...	...	...	...	...	...	...	...	1.80-	2.10	1.95	77,000	60,000	28	16	V	174,000	130,000	24	14	441	59
Silico manganese steels	9260	0.55	0.65	0.60	0.50	0.70	0.60	0.045	0.045	...	...	...	...	...	...	...	...	1.50-	1.80	1.65	78,500	61,000	24	14	V	194,000	145,000	22	13	457	61
Silico manganese steels	..	0.25	0.35	0.30	1.50-	2.00	1.75	0.035	0.035	...	...	...	...	...	...	...	...	0.75-	1.00	0.85	87,000	68,000	22	13	V	211,000	160,000	21	12	470	63

NOTE—The values given above are the average of results which will be obtained in general commercial practice of heat treatments. These values can be increased somewhat when the heat treatments are conducted by laboratory methods.

cretion of the maker in accordance with the size of the part or the purpose for which it is designed.

In Table 2 are listed the physical properties of the S. A. E. steels specified by the Engineering Division. These properties are based on results obtained from actual tests and represent what can be expected when the parts are heat treated according to general commercial methods. The values given can be increased somewhat when laboratory methods are used.

The heat treatments referred to in Table 2 are identical with those given in Table 1, as regards any given treatment and also its modifications; for instance, heat treatment E, E-1 and E-2, Table 1, may be applied instead of heat treatment E, Table 2.

The Engineering Division has had occasion to collect data on a number of tests showing the variation of physical properties of steel with carbon content. Table 3 is based on a large number of laboratory experiments on samples with varying carbon content, the chemical composition being otherwise substantially the same. To secure the required samples, it was necessary to permit a slight variation in the amount of manganese, silicon, etc., from that actually given in the table, but these variations were not sufficient to affect the physical properties. The values of manganese and silicon are considered by the Engineering Division the most desirable ones for the carbon steels referred to in Table 3.

During the recent emergency the enormous demand for steel products naturally tended to result in shortages of such material. The Engineering Division has found it imperative to specify as many kinds of steel as will render satisfactory service. Accordingly the different S. A. E. steels were grouped in such a way as to bring together those with similar physical properties. Then, instead of giving on a drawing one single steel, permitting the use of only one chemical analysis, all the steels in a group have been specified. This group method has been so successful in the motor transport work that it is believed the S. A. E. Standards Committee should classify the various steels according to elastic limit and percentage reduction of area. Such an activity would undoubtedly be of great value to the whole automotive industry.

Standard Instructions Book

As in all other large drafting-rooms, it was early found essential to establish standard practices, not only for the methods of preparing drawings and specifying fits and limits but also as regards the use of small parts, bolts, screws, nuts, washers and rivets, that can ordinarily be purchased in the open market. The Design Section has taken care of this by a "Standard Instructions" book which is made up in loose-leaf form of $8\frac{1}{2}$ by 11 in. blue-print sheets. These sheets were prepared by a standards squad and copies of the book were furnished to all the squad chiefs of the Design Section to obtain uniform practice. To make doubly sure of this, every group of drawings for a particular unit has been checked by a checker attached to the squad doing the work. These drawings and the materials list prepared at the same time have then been passed to a head checking squad which approved the work for drafting-room practice, general design, and for the coordination with other parts affected.

The Standard Instructions book consisted of a few sheets only at the beginning but it has been added to from time to time until it now includes about 160 sheets. It contains definite instructions covering all questions that arise in the drafting-room, such as the proper size and slant of letters, location, form and content of notes, method of showing threads in section or when hidden, and cross-hatching for sections of materials. A number of the sheets given in vol. 1 of the S. A. E. Handbook were reproduced for the benefit of the draftsmen, in some cases the information thus obtained being added to. It was found, for example, that in a number of sizes the S. A. E. light lock washers were entirely too heavy, and as a result the Design Section standardized a series of extra-light washers. In many cases commercial practice varies and it was necessary to make detailed investigations to settle upon a satisfactory procedure. For instance, eight different systems of wire and sheet metal gages are in more or less common use. Different draftsmen might specify a given sheet metal according to different gages. It was therefore determined in the work of the Design Section to use one standard gage for specifying each type of metal. Upon attempting to do this it was found that there was no recognized authority as to which particular gage system should be used for such materials as sheet steel, brass, iron wire or steel wire.

The manufacturers of these products and also the trade periodicals interested were consulted and as a result a sheet was completed giving their consensus of opinion as to the gages that should be used for all the common sheet, wire and similar material. The information contained in this sheet is now used in all specifications by the Design Section.

The matter of fits has been given close study by the Design Section and complete instructions issued as to the method of specifying the various classes, such as force, driving, tight, push and running fits. Draftsmen are supplied with complete data for shafts up to 6-in. diameter. The practice has been to hold the size of hole constant for all types of fit, within tolerances or wear of reamers, while the diameter of a shaft is varied according to the type of fit desired. The size of the hole is, therefore, made nominal, for instance a $\frac{3}{8}$-in. hole is called for as 0.375 plus or minus the specified limits, in this case plus 0.0005, minus 0.0005, while the shaft dimensions are above or below nominal according to the fit desired.

It has been found that the dimensions of the A. S. M. E. standard machine screws and S. A. E. cap screws overlap to a great extent. The Design Section, therefore, arbitrarily decided that no screws smaller than $\frac{1}{4}$-in. diameter should be made to S. A. E. standards, while none larger than $\frac{1}{4}$ in. should be made to A. S. M. E. standards. The Design Section also found that there was no existing standard for special heads. It was therefore necessary to create a standard and one of the sheets in the Standard Instructions book contains drawings of fillister, round, flat and other common types of heads. These drawings represent the average of data obtained from manufacturers of the different parts.

Interchangeability of Parts

When the Class B truck design was first started, serious consideration was given to the necessity of maintaining interchangeability of parts. The fact that parts are made in so many different factories requires, of course, a comprehensive system of gaging. This work was undertaken mainly by Government inspectors, but it was of interest to the Design Section in that the latter was required to give instructions on the drawings for securing interchangeability. The most difficult problem was to insure interchangeability of threaded parts, particu-

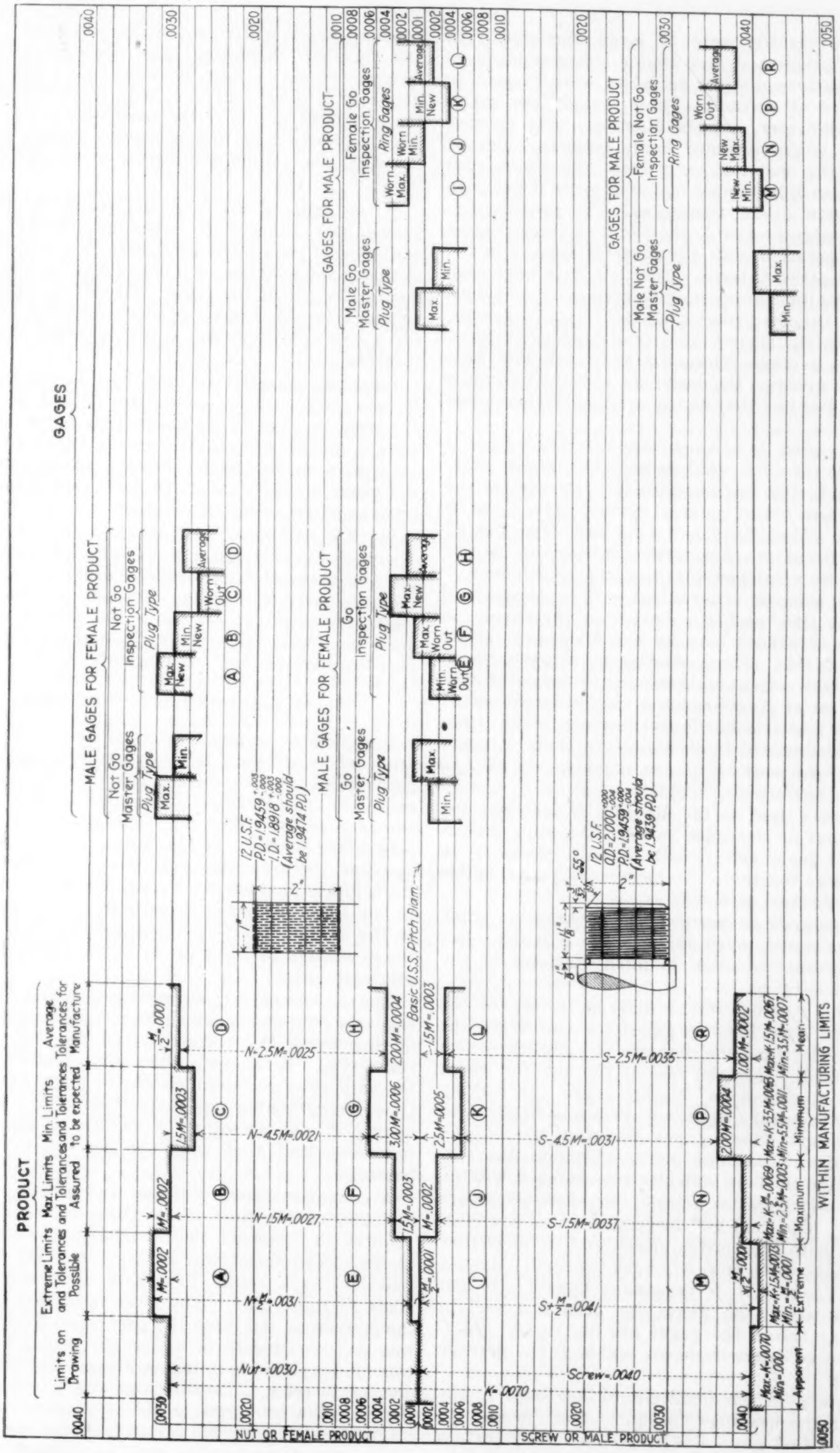


DIAGRAM ILLUSTRATING THREAD LIMITS AND TOLERANCES ALLOWABLE IN THE CONSTRUCTION OF VEHICLES FOR THE MOTOR TRANSPORT CORPS

larly when the threads were of large diameter. An analysis of the problem showed that in commercial practice at least fourteen distinct variables were present in the manufacture of threads. Those include the variation in pitch diameter, outside diameter, base or root diameter, lead angle, included angle between sides of thread and in the shape of their face. To secure production and not unduly hamper manufacturers it has been necessary to allow liberal tolerances. The practice followed by the Design Section is the result of three months of study in cooperation with a number of experts in the manufacture of screw threads. Complete data have been tabulated for the limit dimensions to appear on drawings, these covering threads from $\frac{1}{4}$ up to $7\frac{1}{2}$-in. diameter. In practice with the system of gaging used it has been found that the average tolerances are much less than those originally given on the drawings, that the minimum is considerably less and that on the average the maximum is less.

The gaging system is of considerable interest since it was worked out for what was probably the largest problem in manufacturing interchangeable parts ever solved in this or in any other country, namely, the Class B military truck. The limits and tolerances used on all drawings are based on this system, which can be explained, using the accompanying diagram, as follows:

Suppose that a nut is screwed onto a 2-in. thread and that the nut is then pushed sidewise on the thread so that all of the clearance between the male and female threads is on one side. The accompanying diagram shows the clearances between the male and female threads under each different set of conditions and each different set of tolerances and limits. All dimensions are taken from the basic U. S. standard pitch diameter. This diagram is worked out for a particular case, namely, 2-in. No. 12 U. S. Standard medium fit thread.

In the formulas used on the diagram *M* indicates the minimum amount of clearance between the pitch diameters of perfect male and female threads necessary to permit them to go together. To have something tangible this has been assumed as 0.0002 in.

N is the tolerance or permissible variation in pitch diameter of the screw, taken as 0.003 in. *S* is the tolerance or permissible variation in pitch diameter of the screw, taken as 0.004 in.

The left-hand portion of the diagram refers to the product and the right-hand to gages for measuring the product. To understand the diagram the gaging system must be explained. Take the case of the nut, for which there are "go" and "not go" master gages. In making the gages, it is necessary to permit certain variations for commercial reasons. These gages, therefore, are in each case shown with a maximum and a minimum, the permissible variation in this particular case being 0.0002 in.

With the master gages as a basis, inspection gages are manufactured. The "not go" inspection gages are compared for size with the master gages, and so, of course, the "not go" gage, when new, will have the same diameter as the master gage. It is necessary, however, to get reasonable life from the gages, to permit a certain amount of wear before the gages are destroyed. At *C* a worn-out "not go" gage is shown, and at *D* the average gage condition.

Identical letters, in the small circles, are used in the diagram under the headings "product" and "gages," to indicate the fact that gage *A* is needed to satisfy certain conditions in the product *A*. The letters *A* to *D*, for example, indicate the maximum pitch diameter of the female product which will come within the limits required by the "not go" inspection gage designated by the same letters. In case a new inspection gage of the maximum diameter is used, a product with maximum basic diameter of 0.0032 in. above the basic will pass. If the new gage corresponding to a minimum master gage is used, however, the maximum diameter will be 0.0032 in. above basic.

If the gage in use was just ready for rejection—as shown at *C*—then the maximum diameter of product which could pass would be 0.0027 in. above basic. At *D* is shown the average maximum diameter which will pass during the life of the gage. Master "go" gages are also provided for the female product, and are, in turn, used for a basis of inspection gages to which the product is referred.

Likewise, master "go" and "not go" gages are made for the male product—see letters *L* to *R* in circles—and are used as a basis for inspection gages to which the male product is tested. The calculations for these various conditions can be carried through, as has just been done, the letter on the product in each case referring to the corresponding condition of gage.

In view of the above, a certain set of limits on the drawings—as, for instance, 0.000 to 0.003 in. above basic for female product, and 0.000 to 0.004 in. below basic for the male product—actually gives an average condition in which the tolerance for the nut is 0.0025 in., the tolerances for the screw is 0.0035 in. and the minimum clearance between nut and screw is 0.0007 in.

Changes Required in Design

In an experimental device of any sort changes are required before the final design can be put into quantity production. It may be desirable to change the design originally used in the experimental product or to correct errors revealed either by the experimental work or by actual field service. If the design is fundamentally good, however, the great majority of the changes will be made to facilitate production. Practically all the changes made in the Class B chassis have been of the last type, such as the simplification of castings or forgings, changes in the location of drilled holes or in the limits and tolerances originally set.

The most important difficulty in the first Class B design was due to the rear axle. The trouble here was caused by the worm thrust-bearing, which was of the taper roller type. A complete remedy was found by increasing the number of rolls and the size of rings and at the same time strengthening the drive-shaft and other rear axle parts. It was necessary to do this to take care of the increased load that could be carried by the heavier bearing.

A complete electric lighting and starting system was installed on the first series Class B truck, but at the request of overseas motor transport authorities this electrical equipment was omitted from the second series chassis. As a result, a number of changes had to be made in the design. The most important of these was the method of retaining the generator drive-shaft, which was still required to drive the governor. The generator was first replaced by a tappet plunger, which showed tendencies to seize. The difficulty was finally overcome by the use of a set-screw to hold the governor gear. The original design provided for an overhead shaft connecting the governor mechanism and the butterfly-valve in the inlet manifold. This shaft moved in a tube and was

found to be too flexible, so that it was replaced by a tubing that has worked satisfactorily.

It has been necessary to keep the large number of factories working on military truck parts promptly informed of any changes made in such parts. The system finally worked out for handling drawing changes is most comprehensive. Whenever a change is considered necessary for any reason, information is submitted to the engineer responsible for the particular part or vehicle involved. If he approves of the requested change, the chief of the squad working on the particular vehicle is notified and has the changes executed on the tracings. After the work is done a legend is entered in the right-hand corner of each tracing, this giving sufficient information so that the original form can be restored. Next, a change card is made for all the parts affected, giving the reasons, when the change is to take effect, what disposition is to be made of old stock and by whom the change was authorized. These cards are attached to the proper tracings and the whole thing is checked in the squad by the head checker and then forwarded to the Records Section, which fills out various forms and sends out revised prints. These prints are accompanied by a duplicate copy of the change notice, which enables those receiving the prints to find the change made immediately and also serves as a notification that prints of the same number previously received are obsolete and should be destroyed.

Compilation of Parts Lists

Most of the work done in compiling parts lists has been handled through the Records Section, although it was recently decided to have these lists prepared in the Design Section by the same squad which made the drawings. The parts list system used by the Engineering Division involves a grouping of parts relating to the same assembly in alphabetical order. Any changes in the parts lists are handled in the same manner as are changes on tracings, the necessary notices being prepared and sent out at the same time.

The blueprint apparatus of the Division, which is under the direction of the Records Section, comprises three machines with a capacity of about 50,000 prints a week. These prints are marked with a perforated stamp either for "production" or for "estimate," and are then sent out on properly authorized requisitions.

The activities of the Records Section cover the preparing and keeping up of a number of card files. For instance, an alphabetical record is established by giving the name and other data for each part covered by drawings. Practically the same information is given on a set of cards arranged according to the number of the part, so that the two files are sufficient to determine on what vehicles any given part is used and what effect any change in design will have on these vehicles.

In addition to the supervision of the distribution of blueprints and parts lists, the Records Section has been in charge of the correspondence files of the Engineering Division and of the test reports, catalogs and foreign blueprints. The term "foreign" is used to indicate drawings of other than standardized equipment. The work of the Section has also covered the preparation of all catalogs and instruction books necessary for standardizing motor transport equipment. The repair parts catalog for the Class B chassis, for instance, is a loose-leaf set of sheets giving a numerical list of all parts, a list of the parts according to assembly, and numerous illustrations of the more important parts. After the first 10,000 trucks were completed certain changes in design were made and the repair parts catalog was revised accordingly. Since it was of the loose-leaf type a new page could easily be supplied for any parts changed or added.

"LIBERTY FUEL"

The United States Fuel Administration, which received many requests for information concerning aviation fuel called "Liberty fuel," made public the results of various investigations concerning its composition, the available supply of ingredients, and its efficiency.

Tests made in the aeronautic laboratories of the United States Army and placed at the disposal of the Fuel Administration resulted in information that the fuel was made up of "approximately 65 per cent of benzol; 25 to 30 per cent of kerosene; and the remainder of a small percentage of amyl acetate and probably naphthalene and alcohol, together with, perhaps, small quantities of dissolved solids, and other volatile liquids as yet undetermined."

In this analysis the available supply of the fuel must depend on the amount of benzol, its chief constituent, that is obtainable. Investigation in this direction showed that if the total production of benzol (3500 bbl. a day) were so used, the total production of Liberty fuel would be about 2 per cent of the present output of gasoline, which is approximately 90,000,000 bbl. a year. Benzol, however, has other uses, notably in grease extraction and as a solvent for rubber.

A gumming and corrosion test, also conducted in the aeronautic laboratories of the Army, showed no perceptible corrosion, but "a tremendous amount of gumming; *i. e.*, 0.4 per cent, accompanied by extensive fuming and a penetrating odor." The tendency of the gummy residue in the fuel would be to plug up carbureters, Fuel Administration experts said.

In the same Army laboratory test, crystallization was found to set in at 18 deg. fahr. and to be "practically complete" at 15 deg. fahr. Boiling started at 175 deg. fahr., as against an initial boiling point of not more than 140 deg. for engine gasoline; indicating better starting qualities for the latter.

The Bureau of Standards stated: "The results of tests in an aviation engine indicated that 'Liberty fuel', compared with gasoline fulfilling the export specifications for aviation gasoline, will, when consuming 10 per cent greater weight of fuel per horsepower-hour, develop about 3 per cent greater horsepower. The spark-plug used in 'Liberty fuel' showed a slightly greater carbon deposit than the plugs used in the run of export gasoline."

Information furnished the Fuel Administration by the Navy Department stated that a flight of 40 min. was made with the fuel, and that "no observations were made which show that the fuel would accomplish any other results than to operate the engine in a manner similar to the aviation fuel which had been used." Arrangements were made, the Navy Department reported, for sufficient quantities of the fuel to make full tests, but the fuel was not supplied.

A letter from Benedict Crowell, Assistant Secretary of War, stated:

"The situation has not reached such a stage that this office feels that any definite announcement can be made, or that it can express a definite opinion as to the fuel's true merits."

The Bureau of Mines reported that it was approached, but, on asking for the formula to compound the material, its request was refused. It therefore dropped the matter, declining to act under the circumstances.

CAMP HOLABIRD MOTOR TRUCK REPAIR SHOPS

CAMP HOLABIRD, located at Baltimore, Md., is exclusively a motor transport post, and contains the largest and best equipped repair shop for motor vehicles in the United States or in the world. There is a duplicate under construction at Fort Sam Houston, Texas, and one of 75 per cent capacity in Camp Jessup, Atlanta, Ga. Each of these shops will be the repair center for large geographical districts. Camp Holabird is unique in having a large crating shop, where thousands of motor vehicles have been prepared for shipment to the American forces in France. This work is, of necessity, dependent upon military activities overseas, and comes to an end as soon as the demand for motor equipment for General Pershing's forces ceases.

Camp Holabird is only a little over a year old. In the autumn of 1917, Col. F. S. Leisenring was sent to Baltimore to open a motor transport station, both for repair work and as a shipping base for overseas equipment. Ninety-seven acres of land were bought and construction started, but winter set in early before much headway had been made. Meanwhile the convoying of trucks overland because of lack of railroad facilities began, and frequently large convoys of trucks from the factories arrived and found no parking space except in the streets leading to the camps. Here the men and officers camped, often in the deep snow, while the trucks were awaiting entrance into the parking space. In spite of unheard-of obstacles from weather that delayed arrival of machinery, cranes, tools and men, constant progress was made and by mid-summer the camp had advanced far toward completion, both as regards operation and construction. Today it is a model of its kind, as to its mechanical equipment and its neat and orderly appearance.

Like all the other motor transport bases, Camp Holabird was built for service during the war, after the war and in any future war. It is a permanent institution for the army. The same is true of the mechanical repair units in Atlanta, Fort Sam Houston and El Paso, Texas. All are on Government-owned ground, and have been planned with great foresight.

At Camp Holabird is Mechanical Repair Shop Unit No. 306. It is responsible for the maintenance and repair of all the army motor vehicles from the following States: Maine, New Hampshire, Vermont, Massachusetts, Connecticut, Rhode Island, New York, New Jersey, Pennsylvania, Delaware, Maryland, West Virginia, Virginia, Kentucky, Ohio, Indiana, Michigan and the District of Columbia. The army has within the United States approximately 40,000 motor trucks, 9000 passenger cars, 15,000 bicycles and 6000 ambulances, and it is estimated that 80 per cent of the total are in the area served by Camp Holabird. Thus it will be seen that the demands upon the repair shops are gigantic. All heavy repairs, reconstruction and salvage are taken care of at the Repair Shop Unit. Minor repairs only are made by local repair shops of the service, or by commercial garages.

Repair Shop Procedure

It is a rule of the motor transport repair service that when a vehicle leaves the shop after an overhauling it must be in as good condition for service as when it was new. Motor trucks arrive at the overhaul shop only when major repairs are to be made or units replaced which the local service stations are unable to handle. About 40 per cent of the vehicles coming to a base shop require complete rebuilding; the other 60 per cent require treatment in some more or less important units.

The base shop has a stall capacity for 120 complete overhaul jobs at once. A part of this space is assigned to passenger cars and a part to trucks. Motorcycles also have their respective space. On arrival at the shops, a vehicle is carefully inspected by an officer, who determines what the nature of the repairs will be. The truck is then ticketed and sent in to be washed and assigned to its stall. A crew of men disassemble it; in the case of the complete overhaul, the mechanism is taken to pieces until nothing remains but the bare frame. The various brackets and fastenings of the frame are inspected, loose rivets cut out and redriven, and the process of reassembly is then begun. Meanwhile, the units, as fast as removed from the truck, are ticketed with the truck number and sent to the respective bays in the great shop where such units are handled.

The engine goes to a special disassembling shop, and is given a bath in hot soda to cut the oil and grease from it, and is then rapidly taken to pieces, and passed before skilled inspectors, who examine the condition of each piece, and card it with the work to be done. These pieces then pass to machines whose sole function is to perform the one operation required. On completion of the work according to ticket, the parts once more pass over the inspecting benches, where the quality of the work done is carefully inspected and verified and, if necessary, sent back for readjustment. If acceptable, they pass to the reassembly bench, just beyond. Thus in a few hours' time every part of the engine which is subject to wear has received mechanical treatment that brings it into as perfect condition as when originally manufactured. The original makers employ the same tools that are used in this army shop for repairing and refinishing. When the engine has been reassembled, it is put on the stand and driven by belt power for some hours to smooth the bearings and make the working parts run quietly. It is then removed to a testing stand where it is given gas and run under its own power against a dynamometer, and is not released from this stand until it generates the required amount of power.

While this work is being done on the engine, the transmission is also receiving similar treatment in a separate bay and the axles and differentials in an adjacent bay. Every minor part of the vehicle is repaired by men who do nothing else, and know the last word in tuning up their respective units. In due course of time the units travel to the bay where the original frame stands, repaired and ready to receive them. In the case of standardized vehicles, every unit drops into its place without interference or loss of time. Where a conglomerate mass of different types of vehicles is to be repaired, there is naturally a tremendous slow-down in the rate of repair.

Approximately four working days are required to take a truck completely to pieces and reassemble it, substituting already repaired units in case some particular part of the truck would otherwise delay re-erection. Thus, with 120 stalls fully occupied, there should come out of the factory at least 30 completely rebuilt jobs daily, working one shift. Where the kind of repair is less serious than complete rebuilding, the number would be proportionately

increased. In ordinary work, the shop would probably handle 50 or 60 jobs per day, but under pressure and with double shifts the number could be increased to 100 jobs, of which about 50 would be complete overhauls.

Thus it is seen that this shop is a large institution. It is a specialized plant devoted to doing the kind of repairing which the operation of large fleets of motor vehicles requires. Experience in Mexico and France has taught the army that maintenance is the key to success. A truck literally eats parts, as a mule eats hay and oats. With ample parts available, and repair facilities for putting them in place rapidly, a fleet of motor trucks can accomplish unheard-of things.

The standardized mechanical repair shop unit is 480 by 497 ft. of glass, steel and concrete construction. The personnel required is 55 officers and 1402 men. As a matter of practice the personnel may be more numerous, as it has been necessary to have a considerable number of men in training.

Crating the Trucks for Overseas Shipment

The crating shop, located near the main base shop, is unique. It was constructed at a cost of approximately half a million dollars, with the sole purpose originally of tearing down finished trucks and reducing them to the smallest possible space in crates. Imagine a string of trucks 1½ to 2 miles long, standing on a highway in front of a crating-shop door. As the whistle blows in the morning the first truck is rolled through the door of the shop, and when the whistle blows at night the last truck comes out of the other end done up in a crate 40 in. wide and high and 18 ft. long. Nobody wants a plant for such a purpose except in war times; hence the plant had to be designed for the double purpose of using the same buildings and machinery for a body-building and repairing plant, in which the wooden bodies, sills and special equipment, belonging to a hundred or so types of motor vehicles, can be put together as needed.

In the operation of disassembling a truck for shipment in crate, the truck comes into the plant under its own power, is stopped under an overhead crane, the body is disconnected, lifted and whirled away to a distant corner, where it is torn down and crated. The truck itself is attacked by a squad of men who remove the wheels, put the frame upon a solid wooden platform, which is the base of the crate, and fasten it down with strong bolts. The engine and as many of the major units of the truck as possible, remain undisturbed. Every part of the truck which is detached from the frame has been accurately placed in the crate. It is a fight for inches. Reducing the dimensions of a crate 1 in. in length or in average cross-section means shipping several more trucks per shipload. When the disassembling has progressed to a certain point, the frame, which rests on a bed of rollers, is pushed forward to make room for a new truck and the work of demolition and packing continued a little further on. Another shift, and the finished mass of material is enclosed in prepared side and end pieces. Just before putting the lid on the crate, an oily spray covers every portion of the equipment and leaves the interior of the box enveloped in a mist of oil, as a waterproof cover is nailed on. The finished crate rolls out of the opposite door of the factory in 1½ hr. from the time the truck came in.

Great Number of Standard Vehicles

The necessity for such a complicated repair and construction institution can best be understood when it is known that there are about 100 authorized and standardized types of vehicles required in military work. With complete standardization of chassis, so that these various bodies could be carried upon the minimum number of chassis, this program would require two kinds of passenger cars, four kinds of trucks, one motorcycle, one bicycle, and about eight graduated sizes of trailers. The latter have been found very useful at the front for carrying all kinds of equipment which does not need to be automotive but merely mobile. Machine shops, tanks, sterilizers, water-purification equipment, de-gassing outfits, sanitary outfits for de-lousing clothing, spare parts outfits for carrying the thousand different repair pieces required not only for the trucks themselves but for cannon and small arms, searchlights, radio equipment, heaters for preparing oil and water for airplanes; in fact, there is scarcely an end to the technical necessities which need to be met in order to make a great army quick on the trigger and ready for any emergency.

Such a program is possible only where standardization of the equipment to the greatest possible extent is adopted as the cardinal principle. There was not time in preparing for the present war to thoroughly standardize the motor equipment and the repair program has been enormously complicated and slowed down by reason of this fact. Motor equipment for the army of the future, that is to maintain the prestige which this country has now won by reason of its great effort will undoubtedly be standardized to the last bolt and screw exactly like the rifle which the soldier carries, or the mess kit with which he eats his meals, or the clothes which he wears.

Training School and Supply Depot

Besides the mechanical work accomplished in the base repair and crating shops, Camp Holabird is a large center for motor transport activities. It is the headquarters for the District Motor Transport Officer, from which he administers all motor transport matters in the large area comprising the northeastern district of the country. It is an organization park, at which Motor Transport personnel is received and organized, trained and equipped. These activities will continue in peace times, as the efficient handling of motor transportation requires men who are skilled in the mechanical arts involved, and who also have the discipline and habits of the soldier. This institution will do for the Motor Transport Corps exactly what a recruiting depot or post does for the infantry.

In addition to these activities, Camp Holabird is a supply depot for motor transportation. It will also serve as a central storehouse for the issue of motor truck repair parts and supplies for the army in time of peace. Its enormous fireproof warehouses are capable of storing millions of dollars' worth of parts and tools, and during the next year will be stocked to capacity with the surplus material which was contracted for and in process of shipment at the close of the war.

Within the total tract, now approximately 125 acres, there are gathered some 7000 soldiers, housed in the traditional cantonment buildings, which made famous for efficiency and speed of erection, the Construction Division of the army. These buildings and men constitute a city, which has to be supplied with pure water, sewerage, electric light, streets, sidewalks, hospitals, bakeries, provision warehouses, kitchens, bathing facilities, and lastly, a great auditorium, where the men can be instructed and entertained. Adjacent there are ball and recreation grounds, and a Y. M. C. A. hut. Camp Holabird has a fine band of forty pieces.

Cooling Losses in Internal-Combustion Engines as Affecting Design*

By C. A. Norman†

Illustrated with Charts

THERE is considerable evidence that a great part of the heat loss to jacket water in an explosion engine occurs, not during the expansion stroke but rather in the exhaust passages. Coker found by direct measurement a temperature drop of 150 deg. cent. (270 deg. fahr.) between the gas in the cylinder at the end of the expansion stroke and the gas in the exhaust pipe. Dugald Clerk found indirectly that of a total measured jacket loss of 25.4 per cent, about 16 occurred in the cylinder, and consequently 9.4 per cent in the exhaust passage. Losses in the exhaust passages, however, have no influence on the thermal efficiency of the engine. To arrive at the influence of varying proportions and speeds on this efficiency it would be necessary to determine separately the losses during the expansion stroke for a whole series of engines, varying only one factor at a time. No such investigation has to my knowledge been undertaken.

Clerk, however, has determined very carefully the losses with varying temperature in one engine, and his results combined with other material, most of which will be found reported in his treatise on gas engines, can be made applicable also to other engines. The formula for cooling loss developed later by me from the published work of Clerk and others, while based on purely theoretical considerations, has nevertheless been found to represent fairly both the results of careful experiments and general experience, and its presentation at this time would therefore seem to be abundantly justified.

Experimental Methods Employed by Dugald Clerk

Clerk equipped an ordinary gas engine with a contrivance enabling him to close the inlet and exhaust valves momentarily while the engine continued to run. A series of recompressions and reexpansions of the same burned charge took place in the cylinder. With no cooling and no friction losses the recompression ought always to carry the gas temperature back to its value at the beginning of the previous expansion. As a matter of fact, there is a constant falling of the temperature from the original value. By calculation or determination of the temperature at one point, definite temperature values can be assigned to all the other maximum and minimum-pressure points and an estimate of the actual temperature drops and the average temperatures during the strokes can be made. With the aid of the speed of the engine it is also possible to give the results in terms of reduced temperature drops per second corresponding to certain average temperatures. The results are given in the form of the curves reproduced from Clerk's work in Fig. 1.

These curves show reduced temperature drop per second on the engine running light at 120 r.p.m. for a full expansion stroke (*a*) and for the upper 3/10 expansion stroke (*a'*); also for the engine running loaded at 160 r.p.m. for the full expansion stroke (*b*) and during the upper 3/10 expansion stroke (*b'*). The intersection of the curves with the temperature axis gives the mean temperature of the wall. There are, then, four curves referring to two different ratios of cooling surface to included volume and to four different wall temperatures. It will be noticed that the average wall temperature for the whole stroke is only about 60 deg. cent. (140 deg. fahr.) at light load, and 200 deg. cent. (392 deg. fahr.) at heavy load. At heavy load the average temperature of the combustion space and adjacent parts is as high as 380 deg. cent. (720 deg. fahr.).

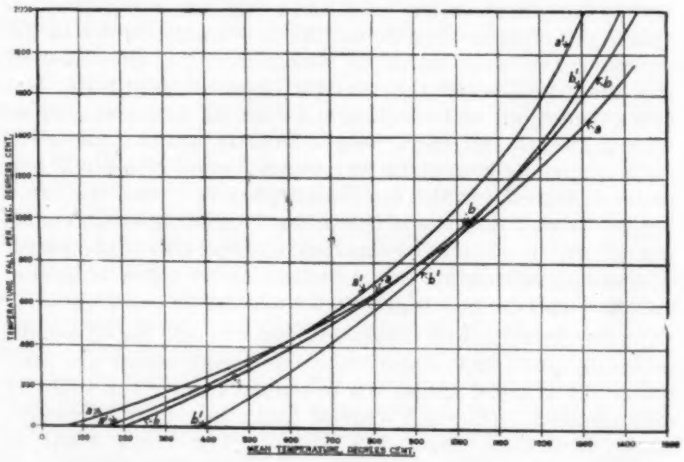


FIG. 1—TEMPERATURE DROP INCURRED PER SECOND AT DIFFERENT MEAN TEMPERATURES CALCULATED IN TIME

Each Line Is the Mean of Three Cards Under the Given Conditions

120 r.p.m. Light { (*a*) Whole stroke; (*a'*) Upper $\frac{3}{10}$ stroke

160 r.p.m. Load 50 b.h.p. { (*b*) Wholestroke; (*b'*) Upper $\frac{3}{10}$ stroke

At the range of average gas temperatures occurring during ordinary full-load conditions, the curve *b* probably represents the cooling losses with all the accuracy to be expected in the present connection. No other curve will henceforth be referred to.

The curves in Fig. 1 give temperature drop not heat loss. The expansion work in an engine cylinder is directly connected with the temperature drop by the specific heat. On the other hand, the work can be directly evaluated from the indicator diagram. In this way the specific heat of the gas in the cylinder at various temperatures can be ascertained. Clerk gives the values found in Table 1, which is a striking illustration of the increase of the specific heat with the temperature. This increase is not uniform, however. It is considerable at lower temperatures, but almost nil at higher ones. Clerk in his original paper deems this behavior partly due to

*From a paper presented at annual meeting of the American Society of Mechanical Engineers, 1918.

†The author is professor of machine design, Ohio State University.

TABLE 1—APPARENT SPECIFIC HEAT (INSTANTANEOUS) AT CONSTANT VOLUME, C_v, EXPRESSED IN FT.-LB. PER CU. FT. OF WORKING FLUID AT 0 DEG. CENT. AND 760 MM.

Temperature, deg. cent.	C_v ft-lb.	Temperature, deg. cent.	C_v ft-lb.
0	19.60	800	26.20
100	20.90	900	26.60
200	22.00	1,000	26.80
300	23.00	1,100	27.00
400	23.90	1,200	27.20
500	24.80	1,300	27.30
600	25.20	1,400	27.35
700	25.70	1,500	27.45

delayed combustion and hence considers his specific heat values as merely "apparent."

Very thorough investigations of the specific heats of gases have been carried out by Nernst and his pupils. From these it would appear that only carbon dioxide behaves with certainty in the manner indicated by Clerk's experiments. The specific heat of all other common gases, even that of superheated steam of sufficiently high temperature, shows rather a linear increase with the temperature. For the mixture used by Clerk the specific-heat values he gives seem to be too high, at least at lower temperatures. This is the opinion also of the Gaseous Explosion Committee of the British Association. Yet, as will be seen below, the deviations are not very great, considering the accuracy possible. By his heat values Clerk has succeeded in accounting very satisfactorily and in a new and striking manner for the heat balances in several combustion engines. In the illustrative examples figured in this paper these values have at times been used. This can be done without question when only general laws for design are to be established.

TABLE 2—SPECIFIC HEAT OF CLERK'S WORKING MIXTURE

Temperature, deg. fahr.	Temperature, deg. cent.	Specific heat, B.t.u. per lb. or cal. per kg. According to Clerk	Specific heat, B.t.u. per lb. or cal. per kg. According to Nernst
Deg. fahr.	Deg. cent.	Clerk	Nernst
32	0	0.1800	0.178
212	100	0.1920	0.183
392	200	0.2020	0.187
572	300	0.2110	0.192
752	400	0.2190	0.196
932	500	0.2280	0.201
1,112	600	0.2310	0.205
1,292	700	0.2360	0.210
1,472	800	0.2400	0.214
1,652	900	0.2440	0.219
1,832	1,000	0.2460	0.223
2,012	1,100	0.2480	0.228
2,192	1,200	0.2500	0.232
2,372	1,300	0.2510	0.237
2,552	1,400	0.2515	0.241
2,732	1,500	0.2520	0.246

The difference between the two series of specific-heat values given in Table 2 amounts to about 10 per cent in the middle of the range, but is barely 1 per cent at atmospheric temperature and less than 2.5 per cent at 1500 deg. cent. (2700 deg. fahr.). With the aid of Clerk's specific heats the cooling curves in Fig. 2 have been obtained.* These curves give the cooling loss during the whole expansion stroke, reduced, however, to heat units per unit weight of gas per second. If, then, we compute the work theoretically obtainable from 1 lb. of gas with the temperatures and the expansion occurring, and if we compute the heat necessary to raise 1 lb. of gas from the temperature at the end of compression to the temperature at the end of the explosion, we can derive the actual expansion work as well as the percentage cooling loss. This, however, applies only to the Clerk engine.

APPLICATION OF CLERK'S HEAT-DROP CURVES TO OTHER ENGINES

Influence of Surface-to-Volume Ratio

Clerk's experimental engine had the following dimensions:

Cylinder diameter, in.	14.00
Stroke, in.	22.00
Total cylinder volume with full-out piston, cu. ft.	2.41
Total cooling surface with full-out piston, sq. ft.	11.20
Cooling surface per cu. ft. of volume, sq. ft.	4.64

The most simple assumption to make with regard to cooling losses is that they vary directly as the ratio of surface to included volume. For this assumption there is considerable support in the results of experiments with explosions in closed vessels. Table 3 gives data on the surface and capacity of the vessels used by several different experimenters, and values of the temperature drop in these vessels as recorded by Clerk.

The values in the last four columns of Table 3 have been plotted in Fig. 3. The experiments were carried out at widely separated places and times by several experimenters, using different methods, with vessels of varying shapes. It should not cause any surprise that the points are scattered. Nevertheless, it can hardly be denied that they are more naturally represented by straight lines, as shown, than in any other manner.

TABLE 3—EXPLOSION EXPERIMENTS WITH CLOSED VESSELS

Ref. No.	Experimenter	Capacity of vessel cu. ft.	Internal surface of vessel sq. ft.	Ratio of surface to volume sq. ft. per cu. ft.	Drop in temperature in 0.05 sec. at mean temperature of 1,450 deg.cent.	1,400 deg.cent.	1,300 deg.cent.	1,150 deg.cent.
1	Hopkinson (large vessel)	6.200	17.30	2.79	80	68	52	38
2	Bairstow and Alexander Hopkinson (small vessel)	0.820	5.02	6.12	127	114	93	65
3	Hopkinson (small vessel)	0.684	4.33	6.33	166	153	131	100
4	Clerk (first vessel)	0.183	1.79	9.78	378	327	257	182
5	Massachusetts Institute Technology	0.180	1.79	9.94	372	327	257	182
6	Clerk (second vessel)	0.150	1.60	10.65	238	216	184	138

As far as experimental evidence goes, it certainly seems to counsel the assumption of direct proportionality between heat loss per unit time and the ratio of surface to included volume. For geometrically similar vessels this ratio varies inversely as the linear dimensions. For similar engines running at the same number of revolutions the cooling losses would then vary inversely as the cylinder diameter. This has led to the conclusion that the cooling losses could be reduced almost to zero by employing sufficiently large dimensions. How unjustified this is, will appear after some further scrutiny.

*The specific-heat values of Clerk have been deduced from work actually done by the gases in his engine. They give, therefore, the heat or energy equivalent of a certain temperature drop in this engine, and should be used in obtaining heat losses from measured temperature drops therein.

Influence of Speed

In the derivation of the heat-loss curve from Clerk's experiments it has been tacitly assumed that the heat loss varies directly as the time. No other assumption with regard to heat transfer has ever been made. With all other conditions equal, the heat loss in an engine should then vary inversely as the speed. It is important however to note that all other conditions should be equal. It is known that if an engine is speeded up something may be gained in efficiency, yet not as much as should be

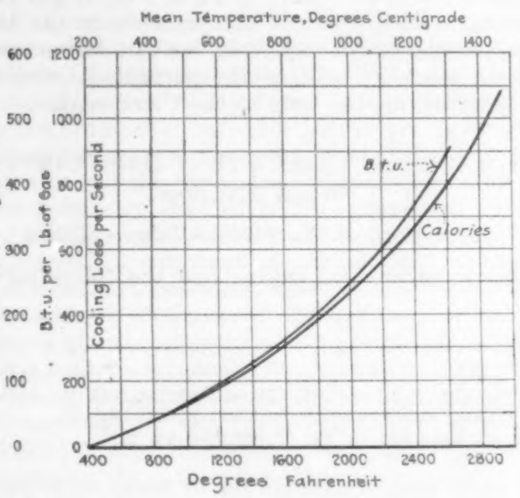


FIG. 2—COOLING LOSS PER SECOND AND UNIT WEIGHT OF GAS FOR FULL STROKE IN CLERK EXPERIMENTAL ENGINE

expected if the heat-loss dropped in direct proportion to the increase in speed. The reason for this is often given as increased turbulence with increased speed. This would of course increase the convection losses. If the valves are correctly located and proportioned there should be no more turbulence at high speed than at low, unless the turbulence is intentionally increased to accelerate combustion.

For the same engine, an increase in speed means an increase in piston speed and an equal increase in gas velocity along the walls. The old idea was that the coefficient of heat transfer from gases increased directly as the square root of the rubbing velocity of the gas. We can probably do no better in the present connection than adhere to this idea, even though it expresses the relations somewhat too simply. We should then put the heat loss directly proportional to the square root of the piston speed. Consequently, if the known heat loss of a certain engine be denoted by L_0, and its surface-to-volume ratio and its piston speed and r.p.m. by R_0, V_0 and N_0, respectively, then for another similar engine with values of R, V and N, working under the same temperature conditions the heat loss is

$$L = L_0 \times \frac{R}{R_0} \times \frac{N_0}{N} \times \sqrt{\frac{V}{V_0}} \qquad (1)$$

This might be checked up on the experiments undertaken by the British Institution of Civil Engineers with three engines of similar proportions but of different size. One of these engines—the largest one—appears to have been identical with the Clerk experimental engine dealt with in this paper. Table 4 gives the main dimensions of these engines, as well as the speed and the brake horsepower developed at the test.

For these three engines Dugald Clerk, who was a member of the investigation committee, gives the following as the most probable values of the indicated horsepower in per cent of total heat supplied:

Engine	L	R	X
I.hp. per cent	31.8	33.7	34.7

By his diagram method he arrives at about 16 per cent as the most probable average value of the cooling loss during the expansion stroke for the X engine. Applying Formula (1) to this value we find the following:

Engine	L	R	X
Figured cooling loss, per cent.	19.9	17.9	(16)
Excess over 16 per cent as figured	3.9	1.9	....
Excess from i.hp., measured..	2.9	1.0	....

TABLE 4—DIMENSIONS OF ENGINES TESTED BY BRITISH INSTITUTION OF CIVIL ENGINEERS

Designation of engine	L	R	X
Cylinder diameter, in.	5.502	9.00	14.008
Stroke, in.	10.000	17.03	22.000
Ratio of stroke to bore	1.820	1.89	1.570
Clearance, per cent of total volume	17.940	18.02	18.590
Surface-to-volume ratio, sq. ft. cu. ft.	10.700	6.52	4.620
B.hp. at test	5.200	20.90	52.700
R.p.m. at test	258.900	203.60	165.800
Piston speed at test, ft. per min...	431.000	577.00	607.000

It must be admitted that the agreement is about as good as could possibly be expected with engines of so widely varying dimensions. The agreement, for that matter, is still better if instead of Clerk's values for the indicated horsepowers we adopt these reported by the committee as a body. These are:

Engine	L	R	X
I.hp., per cent of heat.......	31	32.9	34.8
Difference from value for X..	3.8	1.9	....

The agreement with the figured values is almost exact. Too much importance of course should not be attached to this close agreement, yet, in the absence of other methods of figuring the cooling losses, the use of the simple formula here derived does not seem irrational.

In the form just given the formula is serviceable only

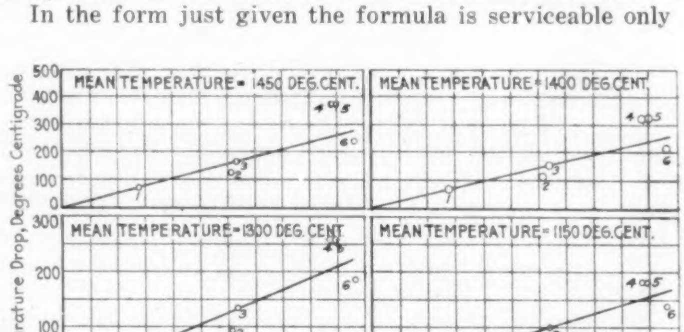


FIG. 3—EXPLOSION EXPERIMENTS WITH CLOSED VESSELS
DROP IN TEMPERATURE IN 0.05 SEC. AT MEAN TEMPERATURES GIVEN PRESSURE BEFORE EXPLOSION, ATMOSPHERIC

for comparing with each other engines working under similar temperature conditions, with similar fuels. It does not take into account variations in expansion ratio or in maximum temperature; nor does it give absolute heat losses in heat units per unit weight of gas. It can easily be given this added range of usefulness by connecting it with the heat losses per second per unit weight of gas given for varying mean temperatures for the Clerk engine by the curve in Fig. 2.

Assuming a certain maximum temperature, or calculating it approximately with the aid of the specific heats given in Table 2, it is possible then to estimate from the expansion ratio of the engine the temperature drop during the expansion by the use of an assumed polytropic exponent. From this we get close enough for our purpose the mean temperature during expansion as an arithmetic average. The corresponding heat loss C is read from the curve in Fig. 2 and it is the loss in B.t.u. per pound or calories per kilogram of gas used for an expansion stroke lasting 1 sec. To get the loss for any other duration of stroke it is simply necessary to multiply C by the actual duration. If the actual speed is N r.p.m., this duration is $30/N$. The surface-to-volume ratio in the Clerk engine was 4.64 sq. ft. per cu. ft., and the piston speed at 160 r.p.m.—the speed at which the curve applies—586 ft. per min. Consequently, for any engine of surface-to-volume ratio R and piston speed V, the expansion cooling loss in heat units per unit weight of gas used is

$$L = C \times \frac{30}{N} \times \frac{R}{4.64} \sqrt{\frac{V}{586}}$$

or

$$L = 0.267(CR/N)\sqrt{V} \qquad (2)$$

In Formula (2) R is expressed in sq. ft. per cu. ft., V in ft. per min., and N in r.p.m., while C can be either in B.t.u. per lb., or cal. per kg., or in any other units for which a C curve is available. In these days of rapid internationalization, with cordial interchange of ideas between France and the English-speaking countries, especially in the field of aeronautical engines, it may be advisable to give the formula also in metric units, as follows:

$$L = 1.14C(R/N)\sqrt{V} \qquad (3)$$

where R is in square meters per cubic meter and V in meters per second, or

$$L = 114(CR/N)\sqrt{V} \qquad (4)$$

where R is in square centimeters per cubic centimeter and V in meters per second.

Finally, if R should be given in square inches per cubic inch, which is more convenient in most cases,

$$L = 3.2(CR/N)\sqrt{V} \qquad (5)$$

where V is the piston speed in feet per minute as is customary.

Application of the Cooling-Loss Formula to Actual Cases

Influence of Absolute Dimensions

Assume in the first instance that the piston speed does not vary. In this case the number of revolutions varies inversely as the diameter, and so does the surface ratio. Consequently for the same C, L is constant. Thus, as far as the formula for the same expansion ratio and the same maximum temperature goes, the cooling losses and hence the indicated efficiency of these engines would be the same for all dimensions. Actually, to avoid cracking of the metal, the cooling will have to be much more efficient for the larger engines. If they are double-acting, then even the pistons will be water-cooled. Even so, the maximum temperatures and maximum pressures will have to be kept down. The larger engines will therefore have a tendency to be rather less efficient than the smaller. On the other hand, with very high values of N the combustion is likely to consume a large part if not the whole of the stroke. In many cases in fact a good deal of the combustion will take place in the exhaust pipe. This can be prevented by extremely efficient carburization and ignition. Yet on the whole the likelihood is that with the same compression the best efficiencies will be found in the middle of the range. This would seem to be borne out by experience.

It may be said however that unvarying piston speed for all dimensions does not correspond with actual conditions. In some airplane and automobile engines piston speeds are found approaching if not exceeding 2000 ft., while it might perhaps be conservative to let the piston speed of very large engines remain below 750 ft. Suppose that at the large-size end the piston speed is half of what it is at the small-size end. The number of revolutions at the former would be only half what it should be to limit the cooling loss to its value at the latter, piston speeds being equal. This would tend to make the cooling loss twice as great at the large-size end. On account of the lower value of the piston speed this loss is reduced in the ratio of $\sqrt{2}$. The upshot is that the loss would be about 40 per cent greater for the large size.

There is then absolutely no reason to look for high efficiency in large dimensions. High speed even though connected with high piston speeds is preferable, even from the pure efficiency point of view—provided the combustion can be effected satisfactorily. Strangely enough, the simple investigations here carried out will give as a sort of by-product some indications as to what it is necessary to assume in regard to combustion.

Influence of Varying Stroke-to-Bore Ratio

Here it will be assummed that the change in stroke-to-bore ratio is not consequent upon a change in expansion ratio. The question is: With a given expansion ratio and temperature range, how should a cylinder be proportioned for best efficiency? A basic cylinder of 5-in. diameter and 6-in. stroke with the same total volume and the same clearance volume for all other proportions, will be considered. The expansion ratio may be 5—that of the Clerk engine; the speed, 1200 r.p.m.; the piston speed will vary from case to case.

Using Formula (1) the conditions found are those recorded in Table 5. The cooling loss increases regularly with increasing stroke-to-bore ratio. The absolute magnitude of the loss is small and the extreme variation changes the thermal efficiency by only 4 per cent. However, this 4 per cent means from 15 to 20 per cent saving in fuel. In an airplane 1 lb. of weight would be saved in fuel for a 10-hr. flight for every horsepower of the engine output. This is not negligible, and will count more and more as longer and longer flights over sea or over enemy territory are attempted.

It might be thought that Table 5 does not really represent the situation, since it refers only to losses during the

TABLE 5—ENGINES WITH UNVARYING VOLUME AND VARYING STROKE-TO-BORE RATIO

Ratio, stroke to bore	1.00	1.20	1.40	1.60	1.80	2.0	2.50
Diameter, in	5.30	5.00	4.74	4.53	4.35	4.2	3.90
Stroke, in	5.30	6.00	6.63	7.25	7.81	8.4	9.75
R, sq. ft. per cu. ft	12.70	12.80	13.10	13.30	13.50	13.8	14.30
Piston speed, ft. per min	1,060.00	1,200.00	1,326.00	1,450.00	1,562.00	1,680.0	1,950.00
Cooling loss, per cent (for Clerk engine 16%)	7.85	8.40	9.05	9.60	10.10	10.8	12.00

whole stroke. With full-out piston the surface-to-volume ratio may be less for large diameters and short strokes. The combustion space however will become more and more disk-shaped as the diameter increases; its surface ratio will be great, and the losses during combustion considerable.

Varying Stroke-to-Bore and Expansion Ratios

The long stroke has in the popular mind become more or less associated with high thermal efficiency. The reason for this may be the high efficiency attained at a foreign test by certain engines having a remarkably long stroke. However, in these engines the long stroke was connected with a very high expansion ratio. The next step is to investigate the influence of varying expansion ratio on engines of the same output and the same speed, the increase in expansion being brought about by lengthening the stroke. With increased compression and increased stroke the output per square inch of piston area will increase for the same number of revolutions. To obtain equal output the cylinder diameter will have to decrease with increased expansion ratio, as indicated in Table 6. In figuring the heat losses in this table it was necessary to use Formulas (2) to (5), since the temperature conditions were no longer the same. The heat losses were obtained in heat units per pound of gas, not as percentages.

TABLE 6—INFLUENCE OF VARYING EXPANSION RATIO ON ENGINES OF THE SAME OUTPUT AND THE SAME SPEED

Volumetric expansion ratio	3.0	5.0	7.0	9.0
Cylinder diameter, in.	5.0	4.0	3.5	3.0
Stroke, in.	5.0	6.0	7.0	7.5
Ratio, stroke to bore	1.0	1.5	2.0	2.5
Surface-to-volume ratio, R, sq. ft. per cu. ft.	13.3	15.5	16.7	19.0
Piston speed, ft. per min	1,000.0	1,200.0	1,400.0	1,500.0
Max. temperature, assumed, deg. fahr	3,270.0	3,270.0	3,270.0	3,270.0
Max. temperature, assumed, deg. cent	1,800.0	1,800.0	1,800.0	1,800.0
Final temperature of expansion, deg. fahr	2,240.0	1,850.0	1,630.0	1,470.0
Final temperature of expansion, deg. cent	1,227.0	1,007.0	887.0	802.0
Heat loss, B.t.u. per lb. of gas	100.0	103.0	108.0	118.0
Heat loss, cal. per kg. of gas	56.0	57.0	60.0	65.0
Indicated work, approximate, B.t.u. per lb. of gas	188.0	248.0	278.0	303.0
Indicated work, approximate, cal. per kg. of gas	104.0	138.0	154.0	168.0

Polytropic exponent in computation of temperature drops, 1.3; compression work assumed equal to 30 per cent of expansion work.

As Table 6 shows, while the cooling loss increases 18 per cent the indicated work increases 61 per cent. The value of high expansion, even though gained by increased stroke-to-bore ratio and decreased cylinder diameters, is hereby clearly demonstrated. For average conditions, however, a high-expansion engine would be more efficient with a large diameter and a short stroke than with a small diameter and a long stroke.

Character of the Combustion in High-Speed Engines

In Table 6 the cooling losses and the indicated work are given in heat units per unit weight of gas. If to obtain percentages an attempt were made to calculate the thermal efficiency of the engine by reference to the heat supplied during combustion, some very astonishing figures would result, such as those in Table 7. Here the maximum temperature has been assumed equal to 3270 deg. fahr. (1800 deg. cent.), in all cases, and the mechanical ciency formulas which have very little to do with actual

TABLE 7—APPARENT SHAFT EFFICIENCIES FOR ENGINES IN TABLE 6

Volumetric expansion ratio	3.000	5.000	7.000	9.000
End temperature of compression, deg. fahr. abs	1,000.000	1,170.000	1,290.000	1,390.000
End temperature of compression, deg. cent. abs	556.000	648.000	718.000	773.000
Temperature rise to 3730 deg. fahr. abs	2,730.000	2,560.000	2,440.000	2,340.000
Temperature rise to 2078 deg. cent. abs	1,517.000	1,425.000	1,355.000	1,300.000
Heat required, approx. ($C_v = 0.225$), B.t.u. per lb. of gas	615.000	576.000	550.000	525.000
Heat required, cal. per. kg. of gas	342.000	320.000	305.000	292.000
Shaft thermal efficiency from indicated work in Table 6 (mech. eff. 0.85)	0.260	0.365	0.430	0.490

It is evident that even though 7 and 9 may be unusual expansion ratios in actual engines, we can never hope to attain brake efficiencies as high as those in Table 7. In arriving at these efficiencies no use has been made of the calculated cooling losses; we have simply taken a polytropic-expansion exponent equal to 1.3 and assumed that all heat is added before the expansion commences to take place. This is the way efficiencies are usually figured, although in combustion-engine practice the procedure is mostly veiled by the employment of mathematical efficiency formulæ which have very little to do with actual processes.

Clerk in his paper before the Royal Society concludes from his experiments on an engine running at only 160 r.p.m. that some combustion is proceeding even during his first re-expansion, *i.e.*, after the whole normal expansion stroke and a whole intervening compression stroke. All experiments with closed vessels show gaseous explosions to take certainly not less than 1/40 sec., and this only with over-rich mixtures. With normal mixtures it takes a much longer time than that to reach the maximum pressure. Turbulence accelerates combustion very much, yet such direct experiments as we have made seem to show that with normal mixtures even a turbulent combustion would take all of 1/40 sec. One-fortieth of a second, however, is exactly the time occupied by the whole expansion stroke of an engine running at 1200 r.p.m.

TABLE 8—SHAFT EFFICIENCY AND COOLING LOSS OF ENGINE OF TABLE 6 WITH EXPANSION RATIO OF 5, ASSUMING ISOTHERMAL EXPANSION

	B.t.u. per lb.	Cal. per kg.
Isothermal expansion work (gas constant = 0.071)	385.000	214.000
Heat added between compression temperature (708 deg. fahr., 375 deg. cent.), and max temp. (C_v = 0.225)	496.000	276.000
Cooling loss, estimated	162.000	90.000
Total heat supplied	1,043.000	5,803.000
Compression work (exponent = 1.3)	104.000	58.000
Indicated work	281.000	156.000
Shaft efficiency (mech. eff. = 0.85)	0.229	0.229
Cooling loss, per cent	15.500	15.500

We have then absolutely no reason to assume that the combustion is complete before the expansion commences. In the series of engines just considered it is far more reasonable to assume that it continues during the whole expansion stroke.

Assuming this, we might approach the conditions actually obtaining by figuring with an isothermal expansion. During such an expansion all the heat added passes directly into work. Assume a temperature of 1600 deg. cent. (2912 deg. fahr.) to obtain during the expansion, then with an expansion ratio of 5 we find for the corresponding engine in Table 6 the values given in Table 8.

The shaft efficiency is now about what would be expected from an engine of this size and speed, and the cooling loss is considerably increased. The main reason for the lowered efficiency, however, is not cooling loss but delayed combustion. To increase efficiency in explosion engines running at very high speeds the main prerequisites are extremely efficient carburetion, extremely efficient ignition, and *perhaps*, if feasible, some means for increasing the turbulence during combustion.

It would hardly be good practice to increase turbulence by greater gas velocities and greater throttling losses in valves. The writer knows of an engine in which very complete combustion is secured by forcing the mixture through narrow grooves in the working piston. The design, however, involves an auxiliary piston driven by an auxiliary linkage, and it is possible that two inlet valves with "clashing" currents might prove of some value. Here a considerable field for invention is yet open.

GASOLINE SPECIFICATION NOTES

THE drawing of specifications for the purchase and sale of petroleum products has not yet been accomplished with any great degree of satisfaction. The analytical methods used for testing gasoline are of fairly recent development and the published descriptions of them are not found in standard textbooks. Some of the methods are decidedly unsatisfactory, having been devised to meet immediate needs when time did not permit thorough study of the factors involved.

There is a lack of knowledge also regarding the practical interpretation of the results obtained in the laboratory, the general tendency being to overemphasize the importance of some particular figure. Specific gravity was formerly considered a complete index of the properties of a gasoline. At present there is danger that overemphasis may be placed on some other equally unimportant factor.

At present little heed need be given to the effect of aromatic hydrocarbons in motor fuel, as these products are in tremendous demand for the manufacture of explosives, dyestuffs and drugs. After the end of the war in Europe it may become necessary to devise a specification clause limiting the percentage of these hydrocarbons that a gasoline may contain, but it is equally possible that such action may be unnecessary, for present indications are that aromatic hydrocarbons are just as good motor fuels as non-aromatic. If limitation should become necessary, it can probably be made most effectively and simply through the specific-gravity requirements. Aromatic hydrocarbons—benzol, toluol and the others—have specific gravities of about 0.87 to 0.88 (29 to 30 deg. Baumé) and boil at the same temperatures as non-aromatics having specific gravities between 0.70 and 0.76 (54 to 70 deg. Baumé).

Distillation Methods

The distillation methods now generally used in the laboratories of refiners vary in some details, but are mostly modifications of the Engler-Ubbelohde method. This method has various basic deficiencies and is inadequate for the analysis of crude oil. It is possible that a modification of the Hempel method might be more satisfactory for general testing purposes. However, the Engler-Ubbelohde methods are so widely employed that they seem as little likely to be abandoned as the Baumé gravity scale. For examining gasoline the method is more nearly adequate than for examining crude oil, because all that is necessary is a uniform basis of comparison for different gasolines.

The use of the various modifications is determined chiefly by the amount of information sought. In some laboratories it is customary to determine only the initial boiling point and the end or dry point, the latter being the highest temperature registered by the thermometer at the end of the distillation. The initial boiling point is variously taken, but generally as the temperature read when the first or the fifth drop falls from the end of the condenser.

The original Engler flask was designed to hold 100 cc. of oil. When only initial boiling-point and end-point determinations are to be made this size is generally used. However, there are certain disadvantages from the use of so small a quantity. When percentage cuts are to be made many laboratories use larger flasks of such dimensions as to give about the same proportionate degree of fractionation as is attained by the original 100 cc. flask.

The American Society for Testing Materials has had occasion to adopt a distillation method for the petroleum products used as turpentine substitutes, and this method, with one or two minor modifications, seems highly desirable for use with gasoline. The details of manipulation are simple and yield satisfactory results. The method involves the use of a thermometer of standard dimensions, a standard 100 cc. Engler flask and an ice-cooled metal condenser tube set at an angle of 75 deg. from the perpendicular. The gasoline is distilled at a standard rate of 4 to 5 cc. a minute, temperature readings being made as each 10 per cent has distilled and when the end or dry point is reached.

It has been customary in the past to read as the initial point the temperature at which the first drop falls. This point is difficult to measure accurately, and there seems to be no advantage in using it instead of the more convenient 10 or 20 per cent mark. It seems that the gasoline should be defined by four points, which can conveniently be the 20 per cent mark, the 50 per cent mark, the 90 per cent mark and the dry point. Some controversy has arisen as to the desirability of a 90 per cent temperature instead of the 95 per cent temperature. The former has the advantage of greater ease of accurate determination. The latter is preferred by some chemists, seemingly on general principles, without very definitely founded evidence. It is thought that by using the 90 per cent temperature and the dry point all needs are met and experimental difficulties are avoided.

Satisfactory figures for these points insure the following desirable qualities of a gasoline:

(a) Enough moderately low-boiling constituents to permit easy starting of engine;

(b) Not enough low-boiling constituents to cause undue tendency toward evaporation losses;

(c) Fairly uniform boiling range and not too great a percentage of constituents boiling at higher temperatures;

(*d*) Only a very small percentage of constituents of very high-boiling point; in other words, not too high an end point.

Requirement *b*, which is satisfied by placing the lower limit for the 20 per cent mark, is introduced for the purpose of making a gasoline reasonably safe and free from evaporation losses. It is a limit that is not likely to be overstepped because of commercial conditions.

—*E. W. Dean* (Bureau of Mines)

National Screw Thread Commission

THE Commission for the Standardization of Screw Threads, authorized by an Act of Congress, was formally appointed by the Secretary of Commerce on Sept. 21, 1918. The duties of the Commission are to ascertain and establish standards for screw threads, which shall be submitted to the Secretary of War, the Secretary of the Navy and the Secretary of Commerce for their approval. These standards, when approved, shall be adopted and used in the manufacturing plants under the control of the War and Navy Departments, and, so far as practicable, in all specifications for screw threads in proposals for manufactured articles, parts or materials to be used under the direction of these departments. It is also intended by the Act that the Secretary of Commerce shall promulgate such standards for use by the public and publish the same as a public document.

Hearings on various topics relating to the standardization of screw threads were held in New York, Washington, Dayton and Detroit, the last hearing in Dayton being completed on Nov. 13. Effort was made to have in attendance at the various hearings manufacturers of screw-thread products and thread cutting tools, large users of both cutting tools and screw-thread products, and others interested in screw-thread standardization.

Discussion of Pitches

Col. E. C. Peck brought out the point that very consistent testimony regarding the adoption of the A. S. M.E. standard thread up to No. 14 had been given at the various hearings.

Mr. James Hartness made the following statement in regard to fine threads: "It strikes me that the 14 thread on the ⅞ and 1 in. diameters is too fine. I would suggest following the S. A. E. system up to and including 9/16, but for ⅝, ¾ and 1 in., having a little coarser thread than the present S. A. E.; ⅞-in. diameter would then be 12 threads per inch. The British standard of 10 threads will probably be used for the 1-in. diameter.

Mr. E. H. Ehrman pointed out that the present S. A. E. system of pitches is as consistent as possible, and is 30 to 50 per cent finer than the U. S. Standard system.

Major O. B. Zimmerman thought the Commission should give good argument for any changes made in present practice in order that the changes should be more readily accepted.

Mr. Wells stated that in the last two years the fine-thread business has increased enormously, as much as 60 to 75 per cent.

Mr. Ehrman said that he could support Mr. Wells' statement from his experience with production; that in product having fastening screw threads and exclusive of such special threads as patch bolts, planer head bolts, boiler bolts, etc., 50 per cent were U. S. Standard and 40 per cent S. A. E. Furthermore, that a few years ago the production was about 40 per cent V-thread, 40 per cent U. S. Standard, and 20 per cent S. A. E. and other threads.

Mr. Hartness inquired of the Chairman whether it would be within the scope of the Commission to suggest anything about an international standard.

The Chairman stated that he thought this would be proper, and that it was important to take steps toward the adoption of an international standard.

Mr. Hartness suggested the 40-in. meter as a means of securing agreement between the English and the metric systems. He stated that 95 per cent of production is to the inch standard, the proportion of production turned out by allied countries being a very small fraction of the total. Furthermore, the change from the present meter of 39.37 to 40 in. would make only about 2 per cent difference in the length of the meter, and this would not interfere with the great mass of engineering work.

With reference to the shifting of 2 per cent, the Chairman called attention to the fact that the really serious and permanent thing of today is scientific work, and that great difficulty would be encountered in getting the scientific man to change to a 40-in. meter.

Colonel Peck suggested that a 10-in. meter be adopted or the present meter be called four meters.

Wire Gages

The Chairman stated that the Commission was not necessarily called upon to consider wire gages, but that he thought it would be desirable to secure a standard.

Colonel Peck stated with reference to wire gages that in the twist-drill business the purchasers had for a number of years specified in decimal fractions, only a very small percentage ordering by number. The sizes given in the drill-rod table are used nine times out of ten.

The Chairman called attention to existing legislation for a sheet-metal gage which was enacted for the Customs service. Unfortunately, this gage is not in common use, and it may be necessary to recommend that it be done away with. The Chairman further stated that the electrical people are in favor of the wire gage based upon area rather than diameter.

Colonel Peck suggested the possibility of having a steel-wire gage based upon diameter and a copper-wire gage based upon area.

Mr. Wells pointed out that the Commission should make definite specifications in order to help the draftsman and purchasing agent, who now use their own ideas, often creating confusion.

Committee on Gages and Methods of Test

The practice of the Ordnance Department in providing "go" and "not go" thread gages for bolt parts and a "not go" gage to check the crest of the male and female threads was said by Colonel Peck to have been successful in securing interchangeability of millions of pieces of ordnance material. Mr. Hartness brought up the matter of the use of the projection lantern for inspecting threaded product as a supplement to the regular gaging procedure.

Classification, Tolerance and Standard Hole Practice

The following basic principles were mentioned by Colonel Peck:

(1) The classification of the various fits of screw threads should be taken up and decided prior to any discussion on allowable tolerances.

(2) Inasmuch as there are many data on allowances for various fits for cylindrical work, these should form the basis of the establishment of the allowances for screw-thread fits when proper interpretation is given to the additional factors affecting the fits in screw threads, namely:

(a) The difference in lead between the external and internal members.

(b) The fact that a screw-thread fit is obtained on the sides of the groove or thread.

(3) The Commission should, if possible, include in its report recommendations for allowances for various fits on cylindrical work, as this will be necessary in the explanations leading up to the decisions with reference to screw-thread fits.

(4) With reference to the matter of tolerances, the practice of the Ordnance Department in specifying working tolerances on screw threads is such that the high and low limits are points in no case to be exceeded. All manufacturing tolerances for master gages, inspection gages and working gages should be accounted for between the high and low limits on the work, in order that these limits should be set as boundary lines over which there could be no dispute. It was pointed out by Colonel Peck that this scheme had been used by the Ordnance Department with great success in the production of millions of pieces of artillery ammunition, with a result that interchangeability was secured with the use of suitable gages, and under widely varying manufacturing conditions, and often with men of very limited training.

Col. Peck's View on Standard Hole

Colonel Peck said "standard hole" practice, as applied to screw threads, literally means a *fixed diameter* of hole for each size, the dimensions of each size hole to be governed by the agreed tolerance required by the commercial tap makers. To illustrate: If tap makers agreed that a pitch-diameter tolerance on a 1-in., eight-thread tap was 0.002 in., then acceptable taps 1 by 8 in. would measure on the pitch diameter between 0.919 in., the basic pitch diameter, and 0.921 in., the allowed limit. A first-class tap of these dimensions would produce a standard hole, and all other taps would be treated as specials, or rejected tools, as in any other well-regulated enterprise. It would be optional with the tap maker whether he salvaged the rejected taps by removing his name and selling them as seconds, or scrapped them; both of these methods are followed by other tool makers, who work to specified tolerances.

The classification of fit would then be obtained by cutting the internal member to suit the allowance and tolerance made for that particular class.

Opponents of the standard-hole practice will say that the method mentioned does not provide enough wear for the tap, but in reality this has nothing to do with it, as the method does not lay down any law preventing the wearing of the tap below basic size if the user is working under a classification of fit with an allowance great enough for the desired wear.

As regards perfect interchangeability, this requires large allowances or small amount of tap wear and tolerances in the work.

The great advantage of standard-hole practice is that taps of known and practically uniform size can be purchased from stock, and only one size for classification of fits need be carried in stock both at jobbers and in tool rooms. All dies are adjustable, and classification of fits desired are made to standard gages with an assurance of proper fit.

It is safe to say that more screws are made by the users and consumers than by the screw manufacturers; therefore, the most good would be done where most needed. Taps are harder to make, and require more time to procure; hence the tap should be the standard member. Further, it is believed that 90 per cent of the work will fall in the classification of fit known as commercial practice; therefore, this entails no hardship on screw manufacturers, as any enterprise is fortunate that can stock 90 per cent of its regular product.

It will be contended that basic size for minimum tap will be a hardship for the die when working on oversize hot-rolled stock. In this standardized age there is no need for this stock to be oversize; rolling mills can on demand roll to size without extra charge.

To offset this, if a standard-size bolt is cut over basic size the thread crest will either not go in a standard reamed hole, or the thread will be truncated more than the standard U. S. form. However, the standard-hole practice makes a strong appeal, and these differences can be reconciled.

Mr. Ehrman's View

Mr. Ehrman said that inasmuch as a great majority of conditions affecting "cylindrical hole and shaft" work are entirely different from those affecting external and internal thread work, it is a serious question whether it is advisable to apply the practice of the former to such an extent as to affect the general present practice, which is to make *basic* a maximum screw size for all except wrench fits.

The tap is not in the same class, as a tool, with the reamer, the accuracy in its manufacture being in terms of thousandths rather than ten thousandths of an inch. The reamer is a sizing tool only, having relatively little metal to remove; hence the size of the hole can be kept within very close limits, in which the extreme accuracy with which it is possible to make cylindrical gages helps very materially. With perhaps one or two exceptions, there is in the case of the hole-shaft problem no factor to take into account other than the tools. In the one exception that comes to mind, commercial shafting, which is a "stock" material, is basic, and the bores of pulleys, hangers, etc., vary according to the class of fit desired. In the screw, bolt and nut industries the "stock" phase is in all probability a far more important factor than is the threading tool or thread gage. In fact, the comparative inaccuracies of the latter with respect to those obtaining in the reamer and cylindrical gage are a strong counter factor for the external screw thread as a basic standard.

For instance, screws, not bolts, form roughly about a half of the external-threaded stock product of this country. As probably from 500 to 1000 screws are used to each tap, it is much more simple to secure quality of fit through the selection of a proper grade of tap which is easily identified by its marking, than to keep on hand different grades of screws, with the danger of their intermixture. In other words, to avoid carrying two grades of taps identical in all respects except the thread limits, the user would have to duplicate his stock of screws, which may include six or seven types, having a range of six to ten lengths for each size and type.

If the maximum screw size (except for force fits) is other than basic, and is variable to secure quality of fit, the *basic* becomes a variable, because taps will continue to be selected or allowed to wear below basic to secure the fit desired. This practice has resulted in the past, and without doubt would in the future, result in loss of interchangeability.

Other very important reasons for making the maximum screw size basic are the following:

(a) To conserve the strength of the screw by keeping the core diameter as large as possible.

(b) To avoid needless expense and restricted output in manufacture, which are incurred if the crest diameters are made below basic size more than required for toler-

ance. This applies not alone to screws in which the bodies or shanks are machined, but to so-called rough bolts, studs, threaded rods, etc., in threading which the die should cut the full size, thus removing the minimum amount of metal from the thread crests and reducing to a minimum the liability of torn threads.

(c) This principle obtains in

1 B. S. W. system
2 B. S. F. system
3 B. A. S. system
4 British threads other than constructional bolts
5 A. S. M. E. machine screw standard
6 Much of our own government work
7 The general practice of screw makers in the manufacture of regular goods
8 General practice of customers ordering screws to their own specifications

(d) Greater simplicity and economy in gages. Each specific size in a screw requires both a thread gage (template) and a set plug, whereas a nut requires one gage—a plug.

(e) The size of an external thread can be more accurately made and more closely maintained to a specific dimension than can that of an internal thread. It can also be more readily and more accurately measured than an internal thread.

(f) In the case of threaded rod, wire-gage sizes, the size of the thread is established by the size of the rod, subject only to the variation due to necessary tolerances in manufacture; the allowance for minimum play or shake being made in the internal thread. Obviously, this practice is necessary to conserve to the fullest extent the strength of the rod. As an example, the practice in the manufacture of bicycle spokes and nipples might be cited. This is similar to that obtaining in general screw practice, placing the tolerance above the basic line and controlling the grade of fit, aside from the influence of the screw-thread tolerance, by varying the size of the tap.

(g) Another most important fact must be taken into consideration: the revision or establishment of standards by other committees and commissions which might feel free to follow the precedent set by this Commission, should it establish the hole as basic, and through the specification of tolerances and allowances, varying from those that may be established by the present Commission, create different basic standards. As the life of screw product may be upward of forty years, it should be clear that any deviation from the present basis of an external thread basic standard cannot but result in certain loss of interchangeability. The experience of manufacturers and users before the elimination of the so-called V-thread standard ought to be a sufficient lesson that deviation from an external thread basic is one of the greatest hindrances to true interchangeability.

STANDARD PIPE THREADS

THE National Screw Thread Commission held a hearing in New York City to obtain testimony from manufacturers and users of pipe threads. Information was presented by thirty-five different manufacturers and users in reply to a questionnaire that had been sent out. One of the topics upon which information was sought was that of threads and these three questions were asked in this connection:

> 1—As a national standard, is there any objection to the adoption of the American Briggs pipe thread sizes for both taper and straight pipe threads as accepted by the American Society of Mechanical Engineers?
>
> 2—In view of the experiments on the form of pipe threads conducted by the Pennsylvania Railroad in connection with the American Society for Testing Materials, which tend to show the desirability of the U. S. Standard form with flat top and bottom ⅛ of the pitch, do you consider it advisable to adopt the U. S. Standard form instead of the present form which specifies a thread depth of 0.8 of the pitch with a resulting flat at the top and bottom of the thread which is quite small?
>
> 3—In your shop practice to what extent do you employ gages for checking pipe threads and what do you consider a satisfactory tolerance for ordinary commercial work stated in turns either way from the gaging notch?

It was the almost unanimous opinion of those present that the Briggs pipe thread sizes should be adopted. In connection with the adoption of the U. S. Standard form, the majority of those present were opposed to making a change. In this discussion practically all of the speakers who opposed the change stated that the present form is satisfactory. One of the speakers pointed out that the U. S. Standard form would not cut into the material as much as the present form but that it would be a rather difficult matter to get a tight joint with it. Another speaker also pointed out that it would not be possible to get pressure-tight joints and that therefore he was opposed to the adoption of the U. S. form. Two other industrial representatives favored the adoption of the U. S. form on account of increased strength of material, while another pointed out that in a competitive test of the Briggs and U. S. Standard form joints 28 per cent more leaks developed with the proposed form, but at the same time 69 per cent more pipe with the Briggs thread was removed and that therefore the proposed form would be more satisfactory. When the U. S. form was first used, the pipe fitters expected the same pressure in connecting the pipe and the fittings as with the Briggs standard and in making a tight joint they did not crush over the top of the thread. The leaky joints were all eliminated by going over them and retightening.

With regard to the checking of the threads by gages and the amount of tolerance that should be allowed, considerable variation in practice was brought out. In some plants gages are used extensively and in others practically not at all. In the case of one company the pipes are threaded to suit the fittings and not gaged, while in another the fittings are threaded to conform to the pipes. In another plant only 10 per cent of the pipes was gaged and the fittings were not checked with gages except when special accuracy was desired. The tolerances given by most of the speakers were either one or one and one-half turns in either direction for both the pipe threads and the fittings, although in one case the tolerance was one turn for the pipe threads and one and one-half turns for the couplings.

Among the firms represented at the hearing were the National Tube Co., Pennsylvania Railroad, Westinghouse Electric & Mfg. Co., General Electric Co., Parkesburg Iron Co., McNab & Harlin Mfg. Co., Malleable Iron Fittings Co., Crane Co., Pratt & Cady Co., Inc., and Pratt & Whitney Co.

Automobile Locking Devices

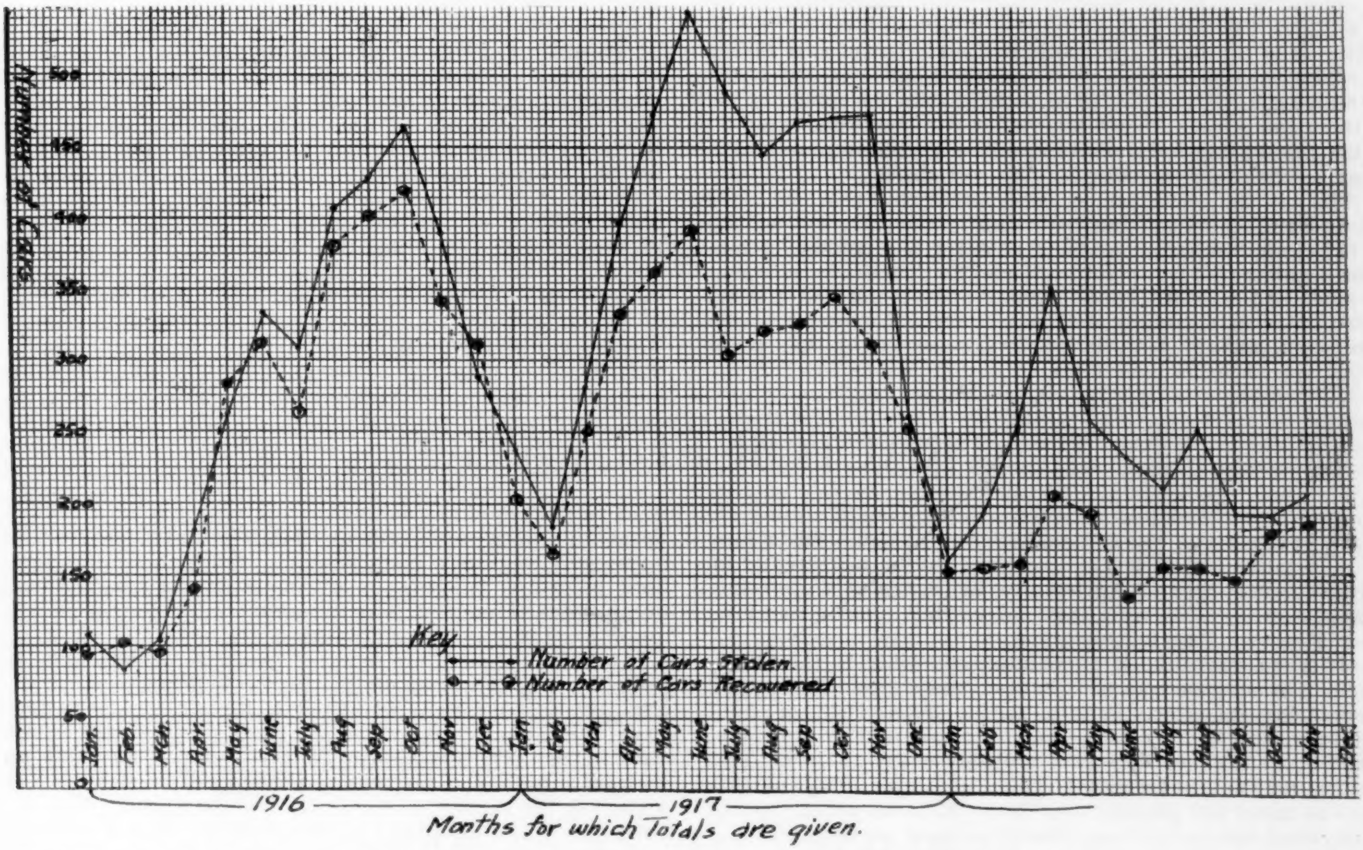
THEFT of automobiles has become so prevalent recently that at the request of the National Automobile Chamber of Commerce, a special committee of the Society was appointed for the purpose of studying automobile locking devices and if possible, recommending some means of combating this evil, as considered from an engineering standpoint. This question is one in which police departments of the various cities are vitally interested as well as the insurance companies and the car owners themselves.

A meeting of the committee was held at Detroit early in December when the subject was discussed from viewpoints of patents, insurance, manufacture, installation as original equipment, value to the car owners and as a practical basis for legislation and enforcement of laws. Different types of locking devices were considered, such as transmission, wheel, ignition, fuel line, etc., as well as heat-treated equipment of heavy construction with internal locks. It was pointed out that one of the greatest values in locks is the time required to open or maliciously destroy them. It was decided that whatever methods of locking are employed provision must be made for moving cars in case of fire or other unavoidable circumstances.

At a subsequent meeting it was brought out that there are a few efficient locking devices on the market at present but that any of these could be opened, destroyed or removed if given time by the men making a business of stealing cars. The relative value of a lock is determined by the time required to manipulate it before a car can be moved under its own power as happens in most cases of theft. From that standpoint it was pointed out that a good transmission lock is probably the best for all cars except Fords, while a good combination ignition and gas lock is effective for that type and second best for all others. For Ford cars a good ignition lock where the holding screws form a part of the circuit and break if the lock is tampered with is the next best arrangement. It was also brought out that steering column locks and wheel spikes are of little value against professional thieves but are useful in preventing "borrowing" of cars by joy riders.

Capt. A. H. Parker of the Auto Squad of the Detroit Police Department stated that the experience of his department showed that a great many thefts were due to the carelessness of car owners in not locking their cars, as required by municipal ordinances. In cities where ordinances are in effect requiring locking devices on cars he suggested that it would be of great assistance to the Police Departments in checking parked cars to see if they are locked, if a plate stating that the car was equipped with a locking device could be required to be displayed in a uniform position on all cars. Another source of trouble is that owners are not able to identify their cars by the assembly, engine, transmission, axle, tire or other numbers, after the car has been stolen or recovered. He suggested that these numbers be checked and recorded by the owner when the car is purchased instead of taking it for granted that they are correct as given in the bill of sale or other certificate.

In describing the experience of the Detroit Police Department in catching and prosecuting automobile thieves and recovering stolen cars, Captain Parker stated that it proved most effective to maintain a separate squad of automobile detectives rather than assign this work to the regular force. In this connection he submitted a record of automobile thefts



MONTHLY TOTALS OF AUTOMOBILES STOLEN AND RECOVERED BY THE AUTO SQUAD OF THE DETROIT POLICE DEPARTMENT FROM JAN. 1, 1916 TO DEC. 1, 1918

RECORD OF AUTOMOBILE THEFTS AND RECOVERIES IN DETROIT

Year	Cars stolen	Cars recovered	Value of cars stolen	Value of cars recovered	Cars recovered, per cent	Per cent of value of cars recovered
1915...	1,391	1,326	....	....	95¼	..
1916...	3,361	3,145	$2,367,304	$2,261,132	93½	95
1917...	4,725	3,569	3,428,358	2,807,046	75½	81½
1918 (11 mos.)	2,425	1,844	2,018,111	1,635,314	76	81

and recoveries from Jan. 1, 1915 to Dec. 1, 1918 which is given in the accompanying table. The chart shows the monthly totals of cars stolen and cars recovered by the Police Department for a period of three years. If the curve of "cars recovered" is above the curve of "cars stolen" it indicates that more cars were recovered than stolen in that one month, some of them having possibly been stolen several months previously. The two general declines in the curves show the falling off in car thefts due partly to the storage of cars through winter months and partly to increases made in the Detroit Auto Squad Force. It will be noted that in the spring of 1918 there was a noticeable falling off in car thefts due largely to the squad being increased to a sufficient force to counteract to some extent the increasing number of thefts.

It was the consensus of opinion that this problem can be best attacked by making the sale of stolen cars unprofitable. The committee recommended that every original sale of a car be accompanied by a "bill of sale" or "deed of ownership" issued and recorded by the factory and provided with spaces for entering transfers of ownership which must be verified before a notary. This should always accompany the car and should be recorded publicly the same as deeds to real estate. A record of these deeds or their numbers should in all cases be filed in the office of the Secretary of State and the car owners or drivers should be required at all times to carry a certificate of registry or the record number of the deed as proof of ownership.

After considering the information obtained it was tentatively recommended that various types of locking devices be regarded as of the following order of value:

1—Transmission locking devices for all cars except Fords and on cars where transmission is located on the rear axle or other accessible place
2—Steering wheel locking devices
3—Combination ignition and gasoline locking devices
4—Ignition locking devices

In all cases it is considered essential that locking devices be operated from the car driver's compartment. The locking devices should be made part of the original construction rather than an accessory to be installed after purchase, and the committee believes that proper classification of devices would enable car manufacturers to put them in as part of the original construction, without trouble and at a minimum additional cost. Where keys are required for the operation of devices it is considered essential that at least 200 key changes for original equipment be provided, the keys to be of the cylinder lock type and milled instead of pressed or filed.

As any recommendations by the committee must be from an engineering standpoint only, it is believed that the National Automobile Chamber of Commerce is the logical organization to consider the commercial and legislative phases of the problem and bring the matter before car manufacturers and state and municipal authorities. It is also thought that efforts to secure uniform legislation and enforcement of laws throughout the various states would be most effective if coming from that organization.

TRACTOR EDUCATION

At the annual meeting of the Tractor and Thresher Department of the National Implement and Vehicle Association Junius F. Cook, assistant to the secretary of the U. S. Department of Agriculture, in charge of farm implement control, delivered a most interesting address on "Tractor Education," as follows:

If a manufacturer is to maintain his place in the tractor field, it is not good enough to have a tractor that is not largely increasing its sales. Reports to the Department of Agriculture show that tractors manufactured in 1916 numbered 29,670; in 1917, 62,742; 1918 probably about 140,000. That is, the 1917 output increased 111 per cent over 1916, and 1918 will be about 120 per cent over 1917. It is not necessary that individual tractor business should increase in just this same proportion, as there are unusual conditions of new tractors coming on the market that in their early days have abnormal increases in sales, due to novelty, unusual advertising and other features used for introducing a new machine. It is, however, necessary that with normal advantages every tractor business should increase, largely depending upon its position in its development. The demand for the different sizes of tractors is varying from year to year. There was the very large machine at an early date; then came a swing over to smaller machines of two, three and four plows. Among many it is thought that the two-plow machine, which in easy going can pull three plows, will be the one most used, even where three and four-plow machines are now being largely used; and I am inclined to this view.

The demand on sizes is narrowing down from year to year, and by large production the smaller sizes will become still cheaper in proportion to the larger sizes than now. The tractor demand as to sizes is today still in a very changing state, and more and more we see that the tractor is bringing changes into farm implement design, and the farm implement modifications are again reacting upon the tractor sizes. Tractor farming is having an influence upon the sizes of farms and also upon farm practice. With these considerations before us it seems to me impossible to be very sure whether the three-plow, the two-plow or the four-plow machines are the ones that will be most desirable.

If a series of sizes are made of the same design, and the two-plow increases sales 50 per cent, the three-plow 25 per cent, and the four-plow 10 per cent, it may not prove very much unless the territory and the total sales of all tractors made in such territory are carefully analyzed. After such an analysis is made and the general tendency of tractor demand is studied, the success of the design may be gaged. Such information is useful, but not conclusive, and points to the fact that after all the figures showing averages of sales, percentages and complaints have been collected and studied, the real success or failure of a tractor concern centers upon the personal training and ability of the manufacturing staff. There may be a fine organization, an energetic and efficient staff, but unless there is a personality that can, and does, put a proper interpretation upon all the data regarding tendencies, and all the information regarding details that are available in the make-up of the machine, it is bound to fall behind. The types of tractors are only partially fixed, and the engineers must not fail to study the type very carefully, as well as sizes and details. The engineer must have a broad view, and make his case so strong that his company will be justified in carrying out his conclusions.

The number of tractors manufactured during the first half of 1918, as ascertained from the tractor questionnaires sent out by the Bureau of Farm Equipment Control, for various horsepower ratings, are as follows: 10-12 hp., inclusive, 2714; 15-16 hp., 3716; 20 hp., 24,128; 22-26 hp., 20,658; 27 hp., 400; 28-30 hp., 2772; 35-36 hp., 1495; 40-50 hp., 1025;

60-80 hp., 1049. These horsepower ratings are those given by the makers, and I believe are not all on the same basis.

The question is always before the manufacturer of how to maintain the high position of his product. The education of his staff that is responsible for the design is most important, and as the tractor business becomes older this education must be given increasing prominence; the older the business, the more important it becomes. There is one other point, and that is the relation between the designer and manufacturer and the user. The point of contact is usually through the salesman, and information is acquired mostly regarding the firm's own machines. It seems to me this source of information should not be depended upon too implicitly or too fully. The salesman's attitude as a salesman is to sell machines, and as such he often brings in useful information and complaints which should receive serious attention. Further than this, close touch with the user entirely independent of the salesmen should be maintained to get a correct, broad view of tractor progress. The manufacturers should be in touch with the tractor schools at colleges of agriculture.

Tractor Education Readily Attainable

The farmer can get valuable education and information from his neighbor's successes and failures, from the tractor schools held by the makers, from the tractor schools at the State Colleges of Agriculture, by attending tractor shows, studying tractor bulletins of the Department of Agriculture and attending any of the tractor demonstrations.

The tractor salesman should, of course, know how to approach his customer, as well as how to describe every part and advantage of the machine. He should also know how to start, run and adjust a machine, as well as what constitutes good work on the farm, such as plowing, harrowing, cultivating, etc.

While the dealer comes in between the manufacturer and the farmer, his education should be such that he knows not only the construction of the machine, but also the principles of design. He must also know how to use it to give the best results to the owner. There is no place in the industry where increased knowledge will reap a richer reward than in the position of the dealer.

Many Users Supply Important Data

From the replies to questionnaires sent out by the Department of Agriculture to farmers, I found that from 2179 reports, the question, "What part of your tractor gives you the most trouble?" was answered as indicated in the accompanying table.

This information gives one a good idea of how to start examining a tractor with the view of buying or studying the machine. It will be seen from this return that one can eliminate at once a great many details and concentrate attention upon the parts listed, which may be considered the main points of trouble.

Magnetos	299
Spark-plugs	110
Gears	108
Carbureter	104
Bearings	80
Cylinders and pistons	61
Clutch	59
Valves and springs	43
Lubrication	29
Starting	28

The farmer must not run away with the idea that because he can make his car do all sorts of things, he can do the same with a tractor running over rough ground and pulling a load up to its full capacity all day. It will take a lot of time and patience to make the tractor man realize that his best load would be two plows; that is, have a margin of one-third for satisfactory running, even if he can just struggle along with three plows.

Manufacturers and dealers will do well not to recommend their machines to take more than two-thirds the maximum power at normal speed for drawbar load.

As it is impossible for every man to become a successful dealer, it is also impossible for every farmer, even if he has a farm suitable for tractor farming, to make a success with a tractor. This should be admitted, and such men should not be encouraged to buy a tractor if the salesman is satisfied it will not be successful. There have been so many letters and requests come to the Department of Agriculture from all sorts of farmers for information to guide them in buying a tractor that I feel there is a real need of the Department taking up this work. Any such work aims at answering the farmers' questions about tractors so far as possible. The tractor could be tested and given a rating such that when a farmer bought a 25-hp. machine he would know such horsepower rating was on the same basis, and would give the same power as another make of machine of the same rating. If such a machine would pull two plows satisfactorily on his farm he would expect a machine having 37 or 38 hp. to pull three plows under similar conditions. He would know what size thresher or ensilage cutter or any other machine he could run with his engine. There seems to be far more need of testing and rating tractors than of motor cars or motor trucks. Such a rating would be a safeguard to the manufacturer and dealer, as well as the farmer, for any dispute arising could be settled by a rather simple test.

SOLDIERS AS ROAD BUILDERS

THE Bureau of Public Roads of the Department of Agriculture has collected figures from the highway departments of the various States indicating the probable number of returned soldiers and sailors that can be employed on road construction and repair work in 1919. Only twenty-nine of the thirty-eight States replying gave definite figures but these indicate that a total of 11,637 skilled laborers and 91,904 unskilled workmen will be needed. The quotas vary all the way from 48 skilled and 100 unskilled laborers in Arizona to 700 skilled and 9300 unskilled laborers in West Virginia. Masons, carpenters, quarry bosses, concrete finishers, road foremen, roller foremen and superintendents are included under the heading of skilled labor but no separation by classes was made in the case of the unskilled workmen.

Petroleum—A Resource Interpretation*

By Chester G. Gilbert and Joseph E. Pogue *Illustrated with* Diagrams

Petroleum is of particular significance because, of all our important resources, it is the most limited and involves the highest percentage of waste. Scarcely one-tenth of the value of the resource is recovered under present circumstances, while the unmined supply available under current practice is only about 70 bbl. to each person. This paper makes an economic study of the resource and the industry engaged in its development and traces the causes of waste to certain maladjustments in the economic situation, pointing out how these can be remedied by a constructive economic policy applied to the matter. The desirability of developing shale oil to replace petroleum as it becomes incapable of meeting the demand is gone into and the advisability of using benzol and alcohol as substitutes for gasoline is considered. The natural gas industry is also treated.

If crude petroleum is exposed to the air, it gradually thickens until a solid residue is left. The first product given off is natural gas; then liquid components evaporate in the order of their lightness; and the final residue is composed largely of either paraffin wax or asphalt. Petroleum is thus seen to be a mixture of different liquids dissolved in one another and holding in solution also natural gas and solid substances.

Commercial quantities of petroleum are found only at depth enclosed within the rocks of the earth's crust. Its occurrence is very similar to that of artesian water, with which, indeed, it is frequently associated. It saturates certain areas of porous rocks, such as beds of sand or sandstone, tending to accumulate where such strata occur beneath denser, impervious layers. Occurring in this way under the pressure that obtains at depth, carrying immense quantities of natural gas in solution, and almost invariably associated with water, petroleum is capable of movement and in general migrates upward until it encounters a layer of impervious rock so disposed in structure as to impede further progress and impound the oil in a reservoir or pool.

Skillful mapping of the surface disposition of rock formations gives the means for determining the structure at depth and hence the position of structural features favorable to the accumulation of oil. When this information is supplemented by careful records of the rock layers encountered as wells are drilled, a three-dimensional knowledge of the earth's crust is obtained, remarkable for its detail and accuracy. Thus by the aid of geological methods the development of petroleum fields may be changed from a gambling venture to an exact science, and, if the scale of operations be sufficiently large, it may be figured rather closely how much oil can be obtained from a given expenditure of money. Instead of representing the most uncertain venture in the world, therefore, oil production can now be made as definitely an engineering project as the mining of a clay bank.

Few questions in geologic theory have been more discussed than the origin of petroleum. It is reasonably certain, however, that petroleum in the main is of organic origin and represents the natural distillation products of plants and animals buried in the muds and oozes of ancient swamps and seas. Vast rock formations, indeed, are known which are nothing more than the accumulated débris of innumerable organisms, compressed, hardened and changed into rock. Fossiliferous limestones, phosphate rock, and coal seams are familiar examples which underlie thousands of square miles of the earth's surface. It would be strange, in fact, if in the process of formation oils were not produced, when organic products today, subject to heat and pressure, yield oily substances not unlike petroleum. Sediments carrying organic remains are sufficiently abundant and widespread to account for all the petroleum that the oil fields of the world give promise of producing.

In spite of an intensive search for new oil regions and vigorous campaigns of development carried on in all parts of the world, the entire supply comes largely from three countries, the United States, Russia and Mexico. Since the beginning of the twentieth century, the rapidly increasing use of petroleum throughout the world has been met largely through the intensive exploitation of American deposits. Thus the United States has assumed a dominant position in respect to this commodity, producing now two-thirds of the world's supply.

The petroleum industry in its ideal form represents a type of industrial activity more highly coordinated than other industries of the present day, affording, therefore, an important object lesson for constructive consideration. In point of fact, however, it is not coordinated throughout, but at present breaks into two portions, by no means in complete adjustment—the production of petroleum and the handling of petroleum with its threefold aspect of transportation, refining and distribution. The conditions of producing crude petroleum are wholly different from those involved in its treatment after it is above ground. This is reflected in the circumstance that over 15,000 individual companies are engaged in the mining of petroleum, while the organizations concerned with the handling of the product are numbered by a few hundred.

Production

Petroleum well drilling is commonly done by a heavy string of tools suspended at the end of a cable and given a churning motion by a walking-beam rocked by a steam engine. This method is known as the standard or percussion system of drilling. The steel tools, falling under their own weight, pulverize the solid rock encountered and literally punch their way to the depth desired. To prevent the caving in of the hole, but especially to avoid the inflow of water from water-bearing formations, the well is lined or "cased" wholly or in part with iron piping, which is inserted in screw-joint sections at intervals during the drilling and forced down to positions needing such protection. The well does not taper, but if deep changes to successively smaller bores at several points, resembling in section a great telescope.

*From U. S. National Museum Bulletin No. 102, Part 6.
†Division of Mineral Technology, U. S. Museum.

Another method of drilling known as the rotary system is also in common use, being particularly adapted to regions where the sides of the well tend to cave badly, as in California and some other localities. This system requires more elaborate machinery than the standard, as the drilling and insertion of the casing are simultaneous. The iron casing, indeed, is tipped with a steel bit and rotated so as to bore its way downward like a great auger. Oil-well drilling is a slow and costly process and makes a heavy draft upon the iron and steel industry, consuming indeed about one-twelfth of its output in ordinary times. The deepest wells are slightly over 7000 ft., but such depths are exceptional. The deepest well in the world is near Clarksburg, W. Va., having recently reached a depth of 7363 ft., according to the U. S. Geological Survey. The cost of drilling in normal times is from $1 to $15 or more per ft. At present, the cost is more than twice the usual figure. To drill a well 3000 ft. deep might now cost from $50,000 to $80,000.

A well favorably located eventually penetrates an oil-bearing bed, and the petroleum may spurt forth in a lavish stream under the influence of the natural gas held in solution under pressure. Such wells are called gushers, and some pour forth prodigious quantities of oil. All wells soon reach a maximum production, after which they pass into a period of decline, and eventually become extinct. So inexorable is this procedure that a curve can be plotted in advance depicting the future behavior of a given group of wells.

The price of crude petroleum at the well varies considerably according to quality, distance from market, and other factors. The paraffin oils of light gravity, such as those produced in Pennsylvania, are the most valuable because they yield the largest percentage of products in demand, while the asphaltic oils of heavy gravity, such as those of California and part of the Gulf region, command a price roughly a fourth of that which the best quality oil brings. Thus the Pennsylvania crude commenced 1915 with a price of about $1.50 per bbl. and ended 1917 at about $3.75, while in the same period California crude climbed from about 35 cents to practically $1. These two types of oil represent the extremes of quality, with the factor of distance from markets nearly the same in the two instances. Between these limits range the prices of all the other oils of the country, the quotation at any given time and location being a complex of quality and of balance between supply and demand, with all the qualifications that the latter expression involves. The wide range in prices for a single raw material, with the utmost concession to differences in location and composition, suggests an undue discrepancy to be credited against the conditions under which oil is produced.

Some crude petroleum is transported in tank cars, but most of the 60,000 tank cars in operation in this country are engaged in moving petroleum products—gasoline, kerosene and fuel oil chiefly. For transportation by sea, steel tankers and towing barges, fitted with noncommunicating compartments, are employed for both crude petroleum and its bulk products. The development of the tank steamer has been an important factor in building up an important foreign trade in petroleum products.

Refining

Crude petroleum may be burned as fuel and nearly a fifth of the domestic consumption is utilized in this way. At the present time petroleum, when completely refined, yields four main products—gasoline, kerosene, fuel oil and lubricating oil—and a large number of by-products, of which benzine, vaseline, paraffin, road oil, asphalt and petroleum coke are well-known examples. Most of these products in turn can be broken up into other substances, each the starting point of further refinements. Under present practice petroleum yields only a few hundred substances of commercial value, but the mind can set absolutely no limit to the number of useful materials that chemical research may still wrest from this raw material.

There are three main types of refineries. The first of these is called a "skimming" or "topping" plant, because the light oils, gasoline and kerosene, are removed from the rest of the products, which are left behind as a residual oil and sold in this semi-crude condition for fuel purposes. The skimming plant, as its name implies, makes an incomplete recovery of products, supplying only those in greatest demand or easiest to make; most of the plants of this kind are situated west of the Mississippi River.

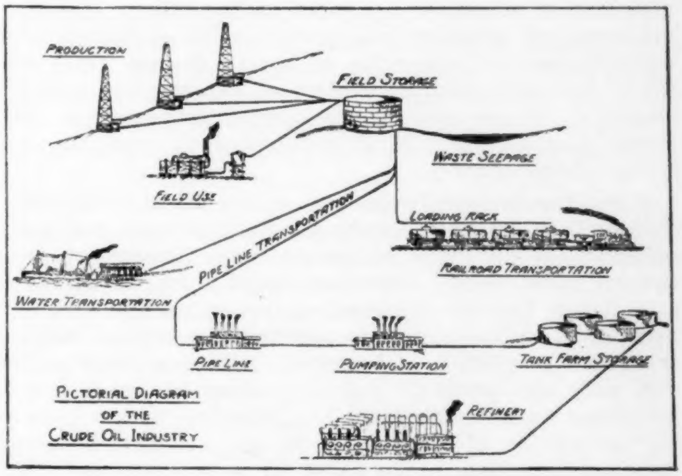


FIG. 1—DIAGRAM OF THE CRUDE PETROLEUM INDUSTRY
From Report of the Committee on Petroleum, California State Council of Defense

The second type of refinery may be termed the "straight-run" plant; this produces all four of the main products, together with by-products, the process separating the crude oil into its natural components with the minimum of chemical change. The straight-run refinery lacks flexibility, because it has no power of producing, for example, more gasoline than the crude oil naturally contains. Such plants are situated in the East and other parts of the country where the demand, especially for lubricants, justifies the expense of the practice.

The third type of refinery is of recent origin, but has made rapid strides toward a great future; it employs the so-called "cracking" process, which yields, like the straight-run plant, a full set of products, but a greater percentage of gasoline than the crude oil gives upon ordinary distillation. This is accomplished at the expense of the heavier component oils, whose molecules are broken into lighter molecules, which constitute just so much additional gasoline. It is obvious that cracking has developed in response to a growing demand for gasoline; its significance is apparent in the fact that it permits the production of a more valuable product from one less valuable. With an increasing call for gasoline and a decreasing supply of petroleum, cracking may be called the hope of the future as regards refinery advance.

Refineries, whatever the type, employ the principle of distillation in their operations. The petroleum is heated in stills and the products vaporize, pass off and are condensed in fractions, representing roughly the materials in demand. These products are then purified by chemical treatment or transformed by chemical means into a series of secondary products. The production of the various kinds of lubricating oils needed for diverse uses represents an intricate, yet single, part of petroleum refining; and is merely one of the many ramifications found in refinery technique. The refining of petroleum makes heavy drafts upon other chemical industries; for example, in normal times about one-tenth of the sulphuric acid produced in the United States goes into petroleum refining, but the refinery in turn contributes many essential products to other chemical manufacturing activities.

While several hundred individual refineries are in operation, the bulk of the output is due to the efforts of less than ten companies. The refining of petroleum, therefore, is largely an integrated activity, in close alliance with transportation of crude, on the one hand, and distribution of refined products on the other. The development of pipe-line transportation has permitted the establishment of refineries at points distant from oil fields, but convenient to centers of consumption and to seaports. Hence one of the largest refineries in the world is at Bayonne, N. J., consuming oil from the interior of the country.

When the famous Drake well struck oil at Oil Creek, Pa., in 1859, an illuminating oil distilled from coal and called "coal oil" was in general use throughout the country. Petroleum, therefore, found a market already established for its illuminating constituent, which it usurped at once, quickly supplanting the coal-oil industry with a production of *kerosene*. Although other products were also produced and lubricating oils made from petroleum found quick favor in connection with a growing application of mechanical energy, kerosene became the chief petroleum product and for over forty years its use grew until this illuminant penetrated literally to the uttermost corners of the globe. It would be difficult, indeed, to estimate the value to the world at large of this cheap and convenient source of light, which has been aptly termed "one of the greatest of all modern agents of civilization." During this period there was little demand for the light products of distillation, the liquids now sold under the commercial name of gasoline, which were, therefore, largely waste products in an economic sense and even in some instances physically destroyed for want of any adequate demand for their utilization. Gasoline for a long time, then, was a by-product of little value turned out in the manufacture of kerosene.

Toward the close of the nineteenth century, however, the commercial application of the incandescent mantle in gas lighting and the development of the electric light introduced a type of illumination so superior to the kerosene lamp in convenience that the use of the latter was gradually relegated in large part to the small town, the country and foreign regions, where the introduction of gas and electricity was not possible. Accordingly, in spite of a most aggressive campaign for foreign trade on the part of the petroleum industry, the refinery faced the restrictions of a slowing demand for kerosene which presaged a limit to the output of the whole set of petroleum products. The menace of this limiting circumstance was destroyed, however, before it became effective, by the introduction and rapid advance of the internal-combustion engine. The phenomenal growth in the use of the automobile built up such a heavy demand for gasoline that this product came into the lead and justified the increasing refinery consumption of crude petroleum, a burden which kerosene, even with the aid of an exporting market for fuel oil, lubricants and other oil products, was scarcely able to sustain. Gasoline is now the main prop to the whole cost structure of petroleum refining.

With the industrial quickening due to the entrance of the United States into the world war, the demand for fuel oil became so insistent that the complexion of the oil situation again changed and the emphasis now fell upon fuel oil, and as the production of crude petroleum did not keep pace with the attempted consumption of fuel oil, a serious shortage of this product resulted, even while the supplies of gasoline were ample to maintain the activities of war, business and pleasure.

If the course of development, as indicated by this broad survey of refinery evolution, be projected into the future, we may reach a time when the petroleum industry will yield a range of fuels for the internal-combustion engine only; illuminating kerosene in quantity narrowing to that desirable for country use and export trade; lubricating oils adjusted to the growing demands of mechanical power; and an ever-widening range of chemical products supporting a great oil by-products industry, rivaling if not exceeding the coal-products industry in importance. In respect to the last, it should be emphasized that the United States faces today an opportunity similar to that which 20 years ago attracted both the United States and Germany as regards the manufacture of dyestuffs, explosives, fertilizers, drugs and other chemicals from the non-fuel components of coal.

Distribution

The entire country is covered by a network of specialized transportation, each step employing a bulk carrier best adapted to its particular purpose both as to size and mechanical facility, the whole involving the maximum of expedition and simplicity. Without this highly organized system, with its far-reaching ramifications, the present widespread use of gasoline and kerosene would not be possible. From the oil field to the consumer, the handling of petroleum is remarkably efficient.

The arrangements whereby a foreign trade has been built up and sustained are no less elaborate. Fleets of tank steamers and freighters carry the products in bulk or in suitable containers to all parts of the world. Fuel oil, gasoline and lubricants go in greater measure to industrial countries, but kerosene penetrates to every corner of the globe, a system of depots and distributing lines adapting the product to the needs of the most out-of-the-way regions. The care that has been bestowed upon the extension of the market for kerosene, against every conceivable obstacle of climate, topography and racial prejudice, is a striking example of industrial foresight; yet without this policy the whole oil industry would have been unable to expand to its present proportions.

The Natural Gas Industry

Natural gas and petroleum are of common origin, the former indeed being merely a volatile component of petroleum, occurring either dissolved in the petroleum under pressure or migrating as the result of the advantageous degree of mobility favoring a gas to positions more or less distant from it. The gas-bearing territory of the United States, therefore, embraces the productive oil fields and a considerable area besides. Natural gas is won in twenty-three states, of which West Virginia, Penn-

sylvania, Ohio and Oklahoma enjoy the largest commercial yields.

The natural gas produced is of two types according to whether it carries a conspicuous burden of gasoline vapor or is lean in this constituent. The first type, as may be surmised, flows from an oil-productive stratum and is called "wet" or casing-head gas, since it makes its appearance from the casing-heads of oil wells. The second type is termed "dry" gas, and comes from portions of porous rock formations practically free from oil; it is produced through gas wells more or less independently of petroleum output.

About one-third of the natural gas consumed in the United States is used for domestic purposes—lighting, cooking and heating—while about two-thirds is burned in industrial plants under steam boilers and especially in metallurgical operations, glass and pottery furnaces and cement kilns, where the requirements of an intense heat call for gaseous fuel. In Ohio and western Pennsylvania, in particular, the abundant occurrence of natural gas has determined the location and widespread development of gas-fired industries.

The natural-gas industry is largely independent of the petroleum industry, although partly overlapping in production. It consists, in the main, of a large number of independent companies in the form of public-service corporations, although some oil companies market their surplus gas. The drilling of gas wells is not essentially different from that of oil wells; but gas, unlike oil, cannot be stored in the field and hence is piped directly to centers of consumption. The gas emerges from new wells under high pressure, but as this declines within a comparatively brief period, the gas field is equipped with compressors which serve to increase the speed and volume of the gas that may be transmitted through gas pipes to distributing stations.

The wastes in natural gas have been appallingly great in the past, and even now, with some of the most glaring points of wastefulness corrected, the resource recovery, by and large, is notoriously small. In connection with the production of oil, especially in fields distant from markets, there has been little incentive to bother with the gas, which has largely been looked upon and treated as a waste product, although now known to be necessary to the proper recovery of the oil. The gas-bearing beds encountered by the oil-seeking drill can be sealed off by mud-laden fluid and the gas conserved for the protection of the oil beds and for subsequent recovery. The gas flowing from the oil-productive stratum along with the oil, particularly in the gusher and youthful period of production, is the casing-head gas from which, since 1910, an increasing production of gasoline has been won.

While the production of gasoline from natural gas has been largely confined to casing-head gas, because of its relative richness in gasoline vapor, the recent development of absorption processes extends the possibility to the bulk of the dry gas produced. It is conservatively estimated that a gasoline production of 100,000,000 gal. per year could probably be attained in this manner, and a significant start is already under way.

The Petroleum Reserve

While unmined petroleum, like other mineral resources not exposed to sight, cannot be inventoried with a nicety of exactness, the proved and prospective oil fields of the United States are, nevertheless, so broadly known that the petroleum reserves can be estimated within a very reasonable margin of error. This has been done by the United States Geological Survey and the accompanying table shows in simplified form the balance sheet as it stands at present:

PETROLEUM RESERVE OF THE UNITED STATES CALCULATED ON A PER CAPITA BASIS*

Rate of production (1917), bbl.	3.4
Mined from 1859-1917, bbl.	42.0
Now underground (1918) and available under present methods of mining	70.0

*The figures given are in round numbers based on a population of 100,000,000 and are calculated from data which take into account the productive possibilities, not only of pools already demonstrated to contain oil, but also of those untested areas in which the geologic evidence is promising.

It is evident from the foregoing table, based on the accumulated knowledge of hundreds of workers in petroleum geology, that an imposing proportion of the petroleum supply is used up. But this table does not tell the whole story; the consumption of petroleum, to say nothing of increase in population, is growing from year to year at a rapid rate, so that a continuation of the present tendency would exhaust the petroleum remaining in an alarmingly short period of time.

This aspect of the situation is depicted graphically in Fig. 2. With no pretense to prophecy, the diagram expresses the situation that faces the country today, and the most generous allowance of margin to cover possible underestimates of future discoveries does not materially change the nature of the issue. A big fraction of the domestic petroleum is gone; whether that fraction is one-third, as present knowledge indicates, or is one-fourth or even one-fifth, makes no difference in the consideration demanded by the situation. The fact remains that the size of the fraction has meaning to people using petroleum today and therefore represents an economic factor that must be reckoned with now.

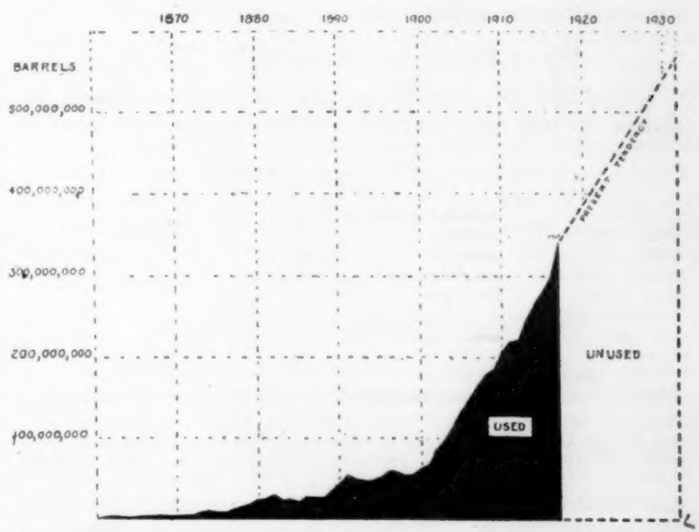


FIG. 2—CHART SHOWING THE PRESENT TENDENCY OF THE UNITED STATES IN RESPECT TO ITS UNMINED RESERVE OF PETROLEUM
DATA FROM U. S. GEOLOGICAL SURVEY

It is, of course, very evident that the present tendency cannot persist to the point of even approximate exhaustion, because conditions naturally arising, such as price increase, growing imports and others will serve to relieve the tension and thus spread the remaining supply over a greater number of years. Hence, in spite of its sensational character, the physical exhaustion of the petroleum resource is a theoretical matter of academic interest

purely. But of practical importance is the period of economic stress to be ushered in when the resource meets a greater demand than it can fill in the customary manner. That will be a period of readjustments to meet new conditions, and will arrive far in advance of physical exhaustion.

As a matter of fact, local adjustments are constantly under way, as petroleum fields reach their climax of production and pass into a period of decline. Thus each field forecasts the history of the resource in its entirety. These local adjustments affect the industry in the way of causing geographic shifts in activities, but they have thus far had no national effect, because new fields have heretofore been ready and able to sustain the shifted burden, but obviously a limit must eventually be reached when an adequate array of fresh fields will be lacking. A consideration of the present situation in this light brings the realization that since a dominant proportion of our petroleum supply is drawn from the Kansas-Oklahoma and California fields, their decline can scarcely be compensated for without the development of other fields to a degree for which there is no prospect. It is generally conceded, too, that these two fields have well-nigh, if not already, reached their productive climax.

It would appear, therefore, that entirely apart from the size of the petroleum reserve, dependency upon a cumulative oil-field development presages a time, soon to arrive if not already here, when the present rate of production can no longer be sustained in its full vigor. Just so soon as the aggregate output is compounded of used and fresh fields, with the latter no longer in the ascendency, the resource as a whole will pass inevitably into a period of slowing and more costly production, even though the resource is yet but half exhausted. The period of economic stress, then, depends merely on this concatenation of circumstances, and by no means upon a marked physical exhaustion of the resource.

What Petroleum Exhaustion Means

It appears from the foregoing that the petroleum resource is not only strictly limited in size but also in

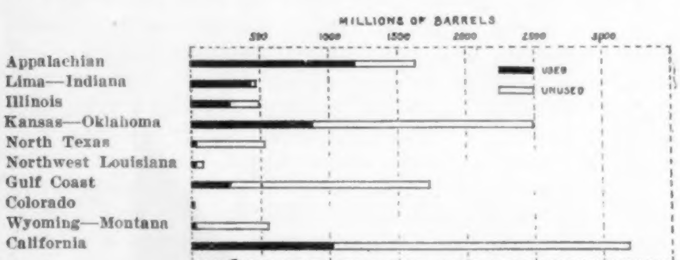


FIG. 3—CHART SHOWING THE APPROXIMATE DEGREE OF EXHAUSTION OF THE PRINCIPAL PETROLEUM FIELDS OF THE UNITED STATES

The Unused Portions Can Be Greatly Increased by Reconstructed Methods of Production. Data from U. S. Geological Survey

ability to sustain the present rate of increase in production. To be better prepared to weigh the gravity of the issue we may inquire, then, what would be involved in a curtailment of activities dependent upon petroleum, since this necessity lies in prospect.

Gasoline is responsible for *the most significant mechanical development of the twentieth century—the internal-combustion engine.* The increasing use of this device for generating power, with its great efficiency and adaptability to small movable units, such as the automobile, has colored the whole face of modern civilization. Because of it, a wholly new type of automotive transportation has grown up, at a time when the long-established methods of cumbersome, coal-energized haulage were beginning to impose a serious restriction upon the growth of industrialism and centralization.

Automotives

The motor truck in 1917 hauled over 60 ton-miles of freight for each person in the United States. Its possibilities are still largely unrealized; its continued extension may be expected to replace largely the short spur line of the railroad; and in connection with the growth of a network of good roads, a country-wide auto-truck utilization will furnish an efficient feeder system to the trunk transportation channels of the country. In respect to the prompt delivery of farm produce, whether to railroads or directly to towns, the motor truck has an exceptionally useful opportunity. The whole problem of food supply, indeed, is closely bound up with the matter of adequate facilities for transportation and an appropriate use of mechanical power, for both of which petroleum products have a tremendous field of unrealized usefulness.

Coming into use at a time when the national food problem had taken on a world-wide aspect, the tractor assumes the utmost present importance, while the future demands an extension in its use such as may be expected to largely relegate to the past the old-fashioned methods of handpower and horsepower tillage. Indeed, upon the increased use of mechanical power on the farm by means of the tractor, the motor truck, the stationary engine, and the automobile—all dependent upon a cheap and adequate supply of motor fuel—the food supply of the future turns. Farm work must be made more agreeable and more efficient if a growing population is to be fed.

The present importance and future significance of the stationary internal-combustion engine, the motor boat and the airplane need scarcely be touched upon here. As to the last, its use in the war has led to such development in flying technique as to justify the expectation that this speedy and mobile agent will soon come into a growing measure of service and use in the affairs of civil life.

Back of the widespread utilization of the internal-combustion engine stands *a great industry engaged in the manufacture of automobiles, motor trucks and tractors.* Starting scarcely two decades ago, this activity has grown until it now represents the third industry, in point of financial value and importance, in the country. In 1917 motor vehicles of all kinds to the number of 1,814,988 are reported to have been manufactured in the United States, having a wholesale value of nearly $1,100,000,000. As can well be appreciated, the *automotive industry,* by virtue of the kind and amount of labor to which it gives employment, of its ramifying sales agencies and extensive advertising, and in turn through its use of steel, aluminum, nickel, rubber, leather, wood and other raw materials, *extends its roots throughout much of the industrial fabric of the country.* This industry is wholly dependent upon the adequacy of petroleum products for continued growth.

It would appear, then, that curtailment in the supply of internal-combustion engine fuel would affect a remarkably wide range of interests. The automobile-owning public, farmers, business activities using motor trucks, and the automotive industry with its ramifications—or expressed in another way, *transportation, food production and a large branch of manufacturing—all have a vital concern in this matter.* Under a waning petroleum sup-

ply, these various activities would suffer a progressive narrowing in scope which would be the antithesis of the continued progress that their importance urges.

Kerosene has brought a cheap light to millions of people the world over. This commonplace substance has been America's greatest gift to the uncultured peoples of the globe. The kerosene lamp still remains the solace of the evening hour among the country folk of this country and the natives of nearly every foreign region. A failing supply would return much of the world to the gloom of the flickering candle, a setback that it is the altruistic duty of this country to circumvent, if consistently possible.

Over half of the petroleum currently produced is used as fuel for steam raising, this portion including the crude petroleum employed for fuel purposes and the fuel oil proper turned out by petroleum refineries. The southwestern portion of the United States is wholly dependent upon fuel oil; Pacific coast shipping and naval activities on both oceans draw much of their energy from this substance; and an increasing number of industrial operations in the Eastern half of the country have been employing this convenient fuel. While the application of fuel oil to steam raising is an economic perversion, for which the penalty is severe, the fact remains that the United States is, for the time being at least, hopelessly entangled in the necessity of prolonging much of this wasteful practice, an unduly forced reduction of which would be fraught with disastrous consequences, particularly for the Southwest. Unfortunately, the swing away from coal in favor of fuel oil is still continuing.

As regards lubricating oils, we are confronted with the fact that the whole mechanical equipment of modern civilization is dependent upon lubricants made from petroleum. For the purpose of reducing friction (*i.e.*, conserving energy) few substitutes are in sight for mineral oils; vegetable and animal oils, although preferable for certain highly specialized purposes, are unsuited for general employment, as they oxidize and thicken with use, and tend to become rancid and attack the metal bearings which they cover. With the passing of petroleum, mineral lubricants will be manufactured from oils distilled from shale and from coals, according to methods in operation today in countries lean in petroleum, as Scotland and Germany. At the present time, in the United States petroleum is produced and manufactured into products far in advance of lubricating needs, which means that the lubricating portion of the resource is being exhausted at a rate dictated by the demand for oil for power generation. Thus the exhaustion of our principal lubricant resource is being accomplished with much greater dispatch than is justified by necessity, since part of the fuel demand could be filled by coal and hydroelectricity not involving a sacrifice of potential lubricants.

An interesting vision opens up as to how a great oil by-products industry, through the values accruing to successive refinements of products, may contribute more than it now does to the expense of petroleum production, relieving to that extent the cost distributed among the products universally used in bulk, such as gasoline. It would seem that a farsighted economic policy, properly directed, might eventually contribute to a lowering cost for internal-combustion engine fuel, just as a proper shaping of coal economics could be made to relieve the focus of expense now exclusively borne by fuel coal; the two conspiring to lower the cost of living.

It would scarcely be too much to say that the whole future of civilization depends upon a continued supply of internal-combustion engine fuel and lubricating oil, while the oil by-products potentiality holds out the prospect of presenting to the world, through the energies of this country, a gift even greater than kerosene has been. It would appear to follow, therefore, that these affairs should not be hampered or curtailed, if in any way the resource or its equivalent can be made to carry the responsibility well into the future. It is a matter of universal concern, then, to inquire if the impending exhaustion of the resource can be circumvented by modern scientific and technical knowledge; and, if so, to ascertain the best procedure whereby this constructive force, as yet not fully used in this country, can be brought into effective action.

The Problem

The enlargement of resource capacity can be brought about in three ways: By prolonging the life of the unused portion of the domestic resource as it is now known; by developing low-grade domestic sources not yet drawn upon; and by building up the use of substitutes, particularly for gasoline, upon which heavy and increasing demands focus.

The supply of petroleum unmined is so limited that the maximum should be obtained from it to prolong its availability. The enlargement of the reserve through the discovery of new oil fields, the elimination of wastes, and the extraction of a greater measure of service from the products represent the lines of progress in sight.

Discovery of New Oil Fields

While much of the oil-bearing territory of the United States is still undrilled, there is no hope that new fields, uncounted in our inventory, may be discovered of sufficient magnitude to modify seriously the estimates given. The reasonableness of this assumption will appear from considering the fact that between 1908 and 1916, during which time the most active exploration campaign in the history of oil development was carried on, the reserve was enlarged by only 1,200,000 bbl., a scant four years' supply at the present rate of consumption. This means that the petroleum resource of the country, like the coal resource, is now fairly accurately measured; and it would be vain to expect a significant increment from an unforeseen direction. Of course, new strikes and oil booms are to be expected, but these will lie for the most part within the area already represented in our measure of the petroleum reserve.

Elimination of Wastes in Production

Far more may be accomplished in the way of enlarging the reserve by the elimination of unnecessary wastes in connection with production. Under present practice, from 90 to 30 per cent of the oil is left underground. Then, of the quantity produced, an appreciable percentage is lost by fire, and a significant portion dissipated by seepage and evaporation due to inadequate storage facilities. On the average, therefore, it is safe to say that *less than 25 per cent of the petroleum underground reaches the pipe line.* If we subtract from this the losses involved in improper and wasteful methods of utilization, the recovery factor becomes perhaps as low as 10 per cent.

Knowledge of petroleum technology is far in advance of its application to oil production, due to the fact that this country is actively engaged in producing such knowledge, but has at the same time provided no adequate machinery for making it of use. Of course part of this advance gets into action where the gain from its applica-

tion accrues to specific interests, but by and large there is marked underconsumption of technological science, the discrepancy being often credited, though with questionable validity, against the difference between theory and practice.

The two greatest wastes connected with oil-well drilling are caused by the harmful infiltration of water from water-bearing strata and the uncontrolled escape of natural gas encountered in the course of drilling. Water is a formidable enemy to oil extraction; as the position of the oil depends, in part, upon a nice equilibrium between oil and water, the undue influx of water into the drill hole means a reduced recovery of oil, if not a total loss of the well; and not only may a single well be completely ruined by inadequate protection against water, but, what is more grave, a whole field of operations may thereby be spoiled. The damage done in the past by water is immeasurable and largely irretrievable, but the danger from water can be controlled by a method of cementation already employed with success in California and Texas, whereby a water-tight band of cement is forced into the space between the well casing and the water-bearing stratum.

Many wells, in sinking, penetrate gas-bearing formations, and in such instances it has usually been customary to suspend operations while the gas escaped into the air, to relieve the pressure against which continued drilling was difficult or impossible. The actual waste of gas due to this has been, first and last, enormous, amounting to billions if not trillions of cubic feet, with a fuel equivalent of millions of tons of coal. Indeed, it would be safe to say that over half the natural gas developed to date has not been made use of, but this physical waste, great as it has been, is of small consequence compared with the waste of the energy represented by the gas pressure, the dissipation of which leads to a reduced and more difficult recovery of the oil. The gas is not only substance, but energy, and represents a force which should be conserved for the sake of gaining a proper petroleum yield later. It is rather interesting that the waste of oil and gas involved in the premature production of gas can be prevented by comparatively simple means; namely, by drilling in a medium of mud-laden fluid which serves to encrust the critical parts of the drill hole, sealing off the formations so that there is no improper escape of gas, and preserving the conduit intact down to the productive stratum.

After oil is struck there are many methods for controlling the output so as to avoid the waste incidental to much of the current practice. The flow can be controlled by rather elaborate mechanical devices, preventing an overproduction; gushers "gone wild" can be capped and brought under subjugation; "blow-outs" may be guarded against and prevented; and losses due to fire, seepage and evaporation largely nullified through adequate development of storage facilities. All these gains will accrue more fully through widespread application of well-known engineering technique already successfully practiced in many instances.

The gas that invariably comes forth along with the oil customarily carries some of the lighter components of the oil itself. These components are recoverable by appropriate methods in the form of a very volatile gasoline, which can be blended with a heavier petroleum distillate to form commercial gasoline. Until a few years ago, the recovery of the gasoline suspended in natural gas was neglected, but now a very significant yield of this so-called "casing-head" gasoline is obtained. The natural gas is made to yield up its gasoline either through compression, which squeezes out the liquid content, or by absorption, which entices it out by using a certain type of oil which later is heated and thus forced to yield up in turn the gasoline absorbed. Even with full gasoline extraction, however, there remains in many fields much more gas than can be consumed by legitimate demand, which necessitates a waste of the surplus, unless it can be cheaply transported to points where demand exists. Possibilities open up in connection with processes of liquefaction, by which the gas can be compressed into reasonable bounds for transportation.

Due chiefly to the decline of pressure upon the escape of natural gas, wells quickly mature and then produce at a declining rate; but recent investigations go to show that even when a well is apparently exhausted its full quota of oil has by no means been extracted. On the contrary, as demonstrated by established practice in Ohio and elsewhere, an additional yield can be forced by compressed air or its antithesis, a vacuum. The more promising of the two methods consists in forcing compressed air down to the porous oil-bearing formation, thus driving the oil to positions reached by pumping wells. The full possibilities of these methods cannot be safely forecast, but they are certainly capable, if widely applied, of increasing by a large percentage the future yield of the country over the estimates made in respect to current practice.

Greater Extraction of Values

The main hope of prolonging the life of the resource lies in the twofold direction of applying improved production technique, as already noted, which will, let us say, double the resource, and of gaining a fuller measure of value from the oil extracted, which is capable of multiplying the resource again by another factor no less great. Improvements in value extraction from the petroleum output will come through the extension and further improvement of cracking methods of distillation; through improvements in the design and efficiency of the internal-combustion engine; through the widening use of the Diesel type of engine, thus gradually deflecting fuel oil from its illegitimate role of a steam-raising understudy to coal, and through a carefully planned program for building up a great oil by-products industry, to give multiplications of value to the portion of oil left after the energy, light and lubricating values are extracted.

The cracking method of petroleum distillation has already been adverted to as representing the most promising means in sight whereby the growing demand for gasoline can be met from a slowing production of crude petroleum. Many such processes have been developed, all giving the same result, namely, a larger yield of gasoline at the expense of heavier components; but two of them stand out with especial prominence. These are the Burton process, for many years in successful practice in the refineries of the Standard group; and the Rittman process, recently developed by the Bureau of Mines and now also established on a commercial basis. The importance of the whole matter can be evaluated by having regard to the fact that at present, even with cracking well established in practice, less than one-half of the petroleum produced is manufactured into products representing an ideal apportionment of the raw material into its components. The production of gasoline can be doubled eventually or even more greatly multiplied without increasing the production of crude petroleum.

With the rigors born of resource limitation—a certain eventuality—and as the automobile passes from a luxury to a necessity, a greater and more universal reach toward

engine efficiency can confidently be counted on. Improvements in engine design, however, will not lie along the single line of gaining more energy from gasoline; the effective use of heavier petroleum distillates, such as kerosene and fuel oil, and of other liquid fuels, such as alcohol, benzol and tar oil, will be planned for and the broad trend of engine development will be toward the character of the resource in its entirety.

In point of bulk nearly three-fourths of the petroleum consumed in the United States goes into the production of power. Of this amount, one-quarter is employed in the form of gasoline as an engine fuel, while three-quarters, in the form of crude petroleum and fuel oil, is used as a convenient substitute for coal, chiefly in firing steam boilers. While the efficiency of the internal-combustion engine is much greater than that of the steam engine, now commonly referred to as wasteful in comparison with more modern types of power generation, the use of the superior principle has thus far been confined in this country almost exclusively to an explosion engine using gasoline, the ordinary automobile engine familiar to all. The fact has generally been ignored in this country that a type of engine, comparable in efficiency with the gasoline engine, but making use of heavy oils (as fuel oil and even crude petroleum), and suitable for power generation on a large as well as a small scale, has for many years been in successful use abroad. This is the so-called Diesel type of engine, which had its beginning as far back as 1893, and "has proved to be, from a thermal standpoint, the most economical heat engine thus far devised, and the one that most nearly approaches theoretical maximum efficiency." The Diesel type of engine offers the means for greatly increasing the power-generating capacity of the petroleum yet to be produced in the United States, in itself alone having the ability to double the energy extraction from the 7,000,000,000 bbl. of petroleum still under ground. But the true significance of the prospect does not appear from the general consideration. In connection with marine service this principle has its richest promise; the advantages of oil over coal for ocean shipping are well known and obvious. If America plans, as she must, on a great expansion in foreign trade and the building up of a substantial merchant marine, she would ignore her most potent point of superiority if she neglected the bearing of the Diesel engine on this matter.

A quotation from a report of the United States Bureau of Mines may be of interest:

> The generally wasteful methods of steam raising in this country must give way to the more efficient methods of fuel utilization that now prevail in Europe, if the United States is to maintain its present position or compete with other countries in the manufacturing industries. With a more conservative use of the Nation's abundant fuel supplies and a better development of the by-product industries, there is no reason why the heavy-oil engine should not materially aid in the more efficient utilization of the fuel resources of the United States.

The use of gas oil, a high-grade fuel oil, in the manufacture of city gas represents a practice largely unjustifiable on the basis of resource economy. In 1915 the amount used for this purpose was about 16,000,000 bbl., or roughly 6 per cent of the domestic petroleum production. With the exception of about one-fifth of the amount, which was employed for making oil gas in the Southwest, where coal is lacking, the bulk of the gas oil was used for carbureting or enriching the luminosity and calorific power of the various types of city gas made from coal, to enable the product to meet standards imposed by municipalities—standards in part a hold-over from the days when the flat-flame use of gas made luminosity a necessary feature. While, broadly speaking, the use of oil in gas manufacturing is a degradation, the practice is not only economically justifiable, but actually desirable, so long as the main outlet for fuel oil is for firing steam boilers, a use still more degraded with the added disadvantage of offering a smaller inducement for refining.

Development of Foreign Sources of Supply

In addition to the domestic production of petroleum, this country, since 1911, has been drawing upon the oil fields of Mexico at an increasing rate, so that in 1917 that country supplied roughly one-tenth of our needs. The pools of Mexico, accessibly situated in the Gulf Coastal Plain, are the richest in the world, and are capable of a much greater annual production than has yet been obtained from them. In the Central American region, in general, there are other promising oil districts, though none as yet is developed in a way comparable with the Mexican deposits. It is not unreasonable to expect that further exploration and development will make available a reserve of oil in Mexico and Central America equal to the total remaining in the United States.

The availability of these deposits is incidental upon many uncertain factors, and obviously it would be unwise to depend upon them and thus offset action regarding the efficient utilization of our own resource. At best, these deposits, and especially those in Mexico, if fully available, and barring international complications, would put off the period of petroleum exhaustion in the United States for a matter of say a couple of decades only; hence they do not lessen the urgency of the domestic issue. Moreover, the high-use employment of these deposits—the output is now used dominantly for fuel purposes—would be to the best interest of Great Britain, the United States, and other countries using the output, not to mention the advantage accruing to the producing republics themselves. Indeed, it would seem, so far as such things can be determined from the outside, that Mexico should take the lead among the republics concerned in developing a policy in regard to petroleum development that would prevent production from exceeding the demand for the high-use products, as this legitimate drain can be expected largely to exhaust the supplies within a generation or two.

Development of Oil Shales

Granted the utmost in the development and use of the remaining supply of petroleum, economic pressure from oil shortage will still be not far distant, and attention turns to sources of supply other than the porous rocks of oil fields thus far exclusively exploited in this country. It is of great significance, therefore, that within the past five years geological explorations on the part of the U. S. Geological Survey have definitely established the existence of vast areas of black shale in Utah, Colorado and Wyoming, much of it capable of yielding upon distillation* around 50 gal. of oil, 3000 cu. ft. of gas, and 17 lb. of ammonium sulphate—the whole constituting an oil reserve aggregating many times the original supply of petroleum.†

*Oil shale is not supposed to contain petroleum, but upon the application of heat, it is thought, organic compounds present react to form an oil resembling petroleum from which can be obtained essentially the same products that petroleum itself yields. But this matter needs further investigation, as it is by no means certain that some oil shales, at least, do not actually contain petroleum as such. In either event, however, shale oil is practically the equivalent of petroleum.

†It is estimated that the oil shales of Colorado alone underlie 1400 sq. miles, with an average aggregate thickness of 53 ft., and are capable of yielding 20,000,000,000 bbl. of oil, an amount approximately twice as great as the original petroleum reserve in this country, together with 300,000,000 tons of ammonium sulphate, valuable as a fertilizer, nearly 900 times the domestic consumption of that substance in 1915.

While these shales have only recently come into notice, a similar resource has for many years been profitably exploited in Scotland, New South Wales and France, where nature has been less bountiful with petroleum, while in Germany the extraction of oil from low-grade coal and other bituminous materials has become a well-established undertaking.‡ The financial success and national importance of the Scottish shale-oil industry is particularly significant, as this activity offers an established technology and a basis of experience for application to the domestic oil-shale matter.

It is apparent from the general situation in respect to petroleum that domestic oil shale may soon come into commercial importance as a producing source of oil.§ As a matter of fact, considerable commercial activity has already commenced looking toward the exploitation of the richer shale areas, especially in the Grand River Valley region of Colorado and near by in Utah. It is evident, however, on the basis of geological occurrence and experience abroad, that a shale-oil industry can come into effective action only as a large-scale engineering procedure, accumulating its profit from a narrow margin made significant by the magnitude of operations. The production of oil from shale, involving ordinary mining operations and a large distillation plant, partakes not at all of the nature of small-unit wildcat drilling by which the petroleum fields are developed.

While the most conspicuous oil-shale areas recorded in this country are in Colorado, Utah and Wyoming, with immediate interest centering around those of Colorado and Utah, other oil shales are found in Nevada, California, Montana, Arizona, Oregon, and in many of the Central and Eastern States—aggregating an immense area and representing *a potential source of oil sufficient to supply this country for hundreds of years.* It is evident, of course, that much of this shale has a prospective interest merely;‖ but there are certain beds overlying shallow coal seams which offer themselves as productive possibilities even under present conditions, as the shale is a waste product, to be removed anyhow in connection with the open-cut mining methods coming into vogue for close-to-the-surface coal seams.

Development of Substitutes

Even with the most efficient use of the remaining supply of petroleum and an appropriate development of shale oil in prospect, the petroleum situation can be additionally improved by a progressive utilization of substitutes for gasoline and fuel oil, so as to give better economic balance by relieving the products upon which the heaviest demands fall. Two substances, benzol and alcohol, are suitable substitutes for gasoline, and offer the advantage of a record of successful use in internal-combustion engines in Europe prior to the war and a marked extension of utilization there under the rigorous conditions of the conflict, while coal and hydroelectricity may be brought to the aid of fuel oil.§§

Benzol is a light liquid, somewhat similar to gasoline in character, obtained at present from the by-product coke oven. The production of benzol in the United States is at present small, owing to the fact that only about a twelfth of the bituminous coal mined is treated for the recovery of by-products. The full utilization of benzol, therefore, must go hand in hand with the development of methods whereby coal will be made to yield a complete measure of usefulness. Indeed, the proper utilization of coal demands a market for benzol as an engine fuel, while the proper shaping of the petroleum resource permits and needs the coming in of benzol as an alternative for gasoline. Thus once more appears an example of how closely the various elements of the fuel situation are connected.

The total capacity toward benzol production possessed by the coal annually produced in the United States is upward of 1,000,000,000 gal.,* which in terms of gasoline represents about one-half of our annual consumption of the latter. Compared with gasoline, benzol yields better efficiency in the internal-combustion engine, but presents a slight disadvantage in respect to use in cold weather. It may be used successfully in the ordinary gasoline engine by admitting a little more air than is customary for gasoline, or by mixing with gasoline.

Alcohol is familiar to every one, and as a fuel offers the advantage that it can be made from organic products which reproduce themselves from year to year, and include vast quantities of materials that ordinarily go to waste.† Unlike the mineral fuels, therefore, it does not constitute a drain upon a reserve fixed in quantity. The difference in effectiveness for engine use between alcohol and gasoline is slight; for, whereas gasoline yields a trifle more power to the gallon, and is easier "starting from the cold," alcohol is safer, cleaner and more pleasant as to exhaust odors. The capacity of this country in respect to alcohol production cannot be closely stated, but if the output of alcoholic beverages is any criterion,¶ existing distilleries, upon conversion, could at once produce fuel alcohol to the extent of millions of gallons, while the substitution of waste products for grain would effect a great economy over the cost of denatured alcohol as made at present. If, in addition, the perplexing legal difficulties that now hedge in such a development could be circumvented, the use of individual manufactories on farms could readily furnish a perpetual supply of engine fuel at little cost, where a cheap fuel would have its most far-reaching social effect by tending to lower the cost of food.

Artificial gas made from coal offers a convenient substitute for gasoline in certain types of stationary internal-combustion engines, while the suction producer plant, with its adaptability to lignite and other low-grade fuels, offers a wide field of usefulness in the partial relief of gasoline, especially for motor boats. Artificial gas may

‡The shale oil of Scotland was of great service to the English Navy in the present war by supplying many oil-bearing ships with fuel, to the relief of transatlantic shipments; while the German oil has proved invaluable to that country in supplementing an inadequate command fo petroleum resources.

§"These shale areas will be developed in time on a basis as safe and sane as that of our coal mines of today. When that time arrives oil prospecting will have passed and the whole complexion of oil producing will change. It will, literally, be oil mining with steam shovels in open pits and glory holes, and, later, tunnels and adits. There will be no lack of oil products for several generations to come, but the true oil fields of today will probably disappear within another generation and be replaced by oil mines." Dorsey Hager, The search for new oil fields in the United States: *Engineering and Mining Journal*, Jan. 5, 1918, page 11.

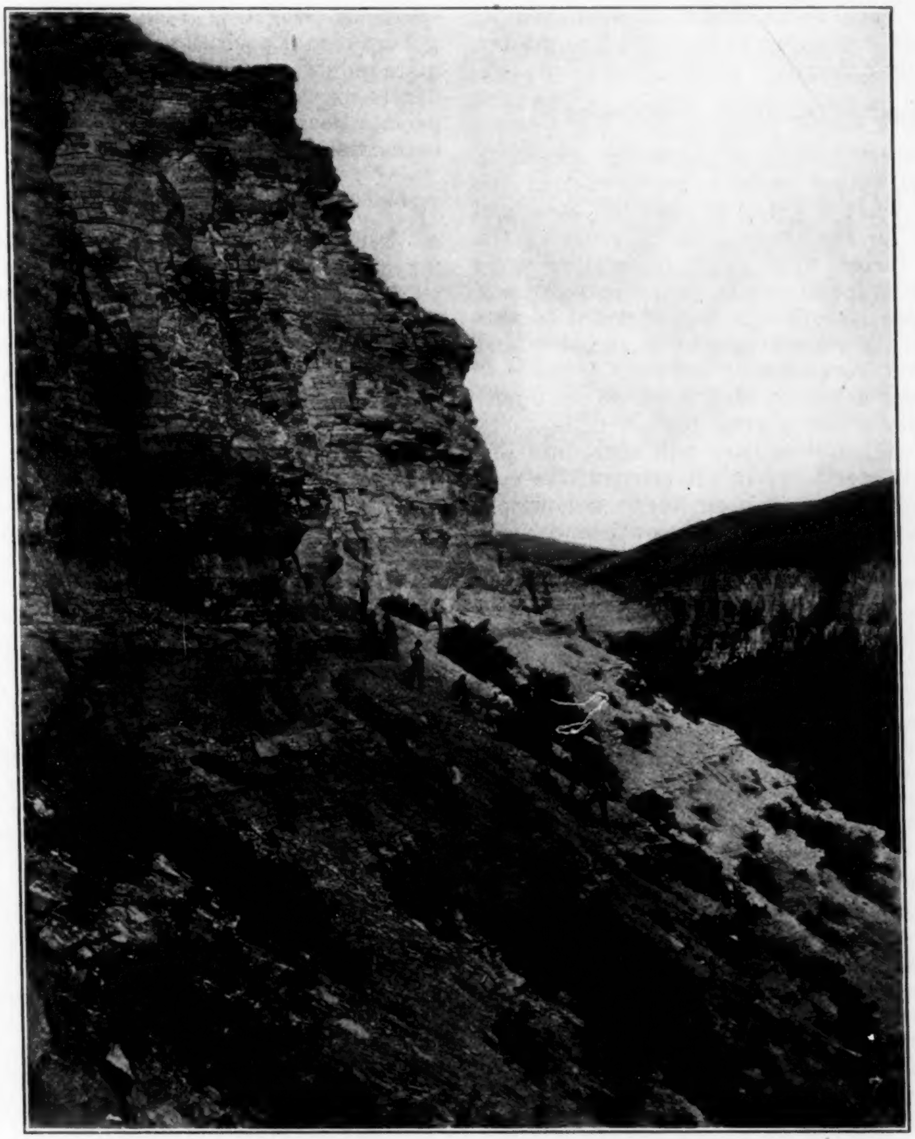
‖Ashley estimates that under present conditions a barrel of crude oil produced from Eastern shale of average quality will cost about $4.20, little more than such an oil would be worth at present, barring by-product possibilities not possessed by its rival, petroleum. Bulletin 641-L U. S. Geological Survey, 1917.

§§Recent work on castor-oil production gives some indication that this organic product may come to be a significant source of engine fuel.

*On the basis of a yield of 2 gal. to the ton of coal.

†Alcohol can be made from starches, sugars, wood waste, sulphite liquors from paper manufacture, peat, cornstalks, etc. Its cost in Germany several years ago was as low as 25 cents per gal., in England 33 cents per gal.—prices comparing favorably with the present cost of gasoline in the United States. Rittman estimated on a pre-war basis that alcohol would become a commercial fuel in the United States when gasoline exceeded 35 cents per gal. *Journal Industrial and Engineering Chemistry*, May, 1917, page 528.

¶It is worthy of note that the consumption of alcoholic beverages and of gasoline during 1916 in the United States was approximately equal; each close to 2,000,000,000 gal., equivalent to a per capita that substance in 1915.



Photograph by courtesy of Denver & Rio Grande Railroad

CLIFFS OF OIL SHALE NEAR GRAND VALLEY, COLO.

even become suitable for automobile use through the development of appropriate methods of compression, or even liquification, so as to enable its storage in small compass. Even without such treatment, but under the stress of gasoline shortage, artificial gas met with successful use in London during the war. Motor buses and other conveyances carrying large canvas containers filled with gas have now become commonplace objects in England.

Fuel oil has come into extensive use in the United States, especially in the far West, as a substitute for coal. It is more convenient than coal, and is therefore adopted by industries wherever its price is low enough to permit such use. Its employment in this way cannot be sustained, in view of the slowing rate of petroleum production and the counter demand that will come from the further development of cracking practice in refining, and from wider adoption of the Diesel type of internal-combustion engine. It will soon be necessary, therefore, in any event, to bring coal and hydroelectric power to the aid of an increasing number of the activities now dependent upon oil fuel; and the whole matter can be facilitated, to the benefit of the petroleum resource in particular, by constructive action in respect to coal and water power, so as to make their service in this respect more immediately available.

The Solution

There are three outstanding features in the petroleum situation of unescapable significance. These are: The strictly limited size and decreasing availability of the petroleum reserve, the increasing importance of certain of the products made from petroleum, and the tremendous waste involved in the current method of bringing petroleum into use. The first two circumstances, of course, make the last important. If there were plenty of petroleum, waste in its use would not matter; or if petroleum were of no great value, merely a luxury, neither waste nor limited quantity would make any great difference. Even if the supply were limited, but sufficient, say, for 50 years, it might be difficult to arouse any interest

at this busy moment in the issue. But petroleum is a basic necessity, as much so as wheat or wool, and its exhaustion is already beginning to be felt. The matter, then, cannot be safely deferred.

Encouragement of Oil-Shale Development

Although the present source of petroleum should be made to render its fullest service, we should at the same time find out what is going to take its place, and prepare in advance for the transition, not ignoring the matter until it is forced upon us by necessity.§ We have seen that oil shale is the only successor in sight, and indeed some attention has already been devoted to this source, especially as the richest part of it occupies land in possession of the Government.

But oil shale being a leaner resource than that now worked for petroleum is not a rival but an understudy to the oil field. A shale-oil industry will come into life when the situation is ready for it. A constructive economic policy will neither force it prematurely nor permit conditions to retard its inception and growth once it is needed to supplement an inadequate oil-field production. The oil-shale matter, then, is merely part of the whole oil problem, and cannot be solved on its own merits. Indeed, it has no merit other than that reflected from a growing scarcity of petroleum. The development will unfold under the influence of an economic necessity for shale oil. A constructive policy may contribute in three ways. It can, in the first place, stabilize the production of petroleum so as to place this resource on the sound basis of ordinary mining procedure. With this done, oil shale will face a resource with which it can cooperate, not an adversary which it must fight. So long as the oil fields of the country fill all the legitimate needs for petroleum products and contribute a large surplus for burning under boilers, there would appear to be no pressing need for shale oil.||

Secondly, the Government can prepare for the time when shale oil will be needed by establishing an experimental plant on a commercial scale, equipped to work out on a practicable basis, with full by-product recovery, the most efficient practice adapted to the conditions of the domestic resource. Such a plant could start with the technique developed in the Scottish shale-oil industry, and by proper research build up a process which would insure the home activity against taking over any obsolescent features of the Scottish practice or from passing through a stage of technological immaturity. This step, too, by providing exact figures of cost would give needed information upon which plans for an industrial development could be made. With the petroleum situation under scientific guidance throughout, and the shale-oil process worked out, the successor to petroleum could come into action on the sound basis of engineering exactness, unencumbered by the speculative element of uncertainty.

In the third place, as most of the richest oil-shale areas are embraced in the public lands, the Government has the responsibility of either itself developing the resource or delegating this duty to private industry. Since governmental operation of matters in the field of legitimate industry is under the ban of public opinion, it is evidently necessary that the resource be made available to private development under terms favorable both to the industrial activities concerned and to the public at large.

Encouragement of Benzol and Alcohol Development

While there is no apparent need at the present moment for gasoline to be relieved of part of its duties by the production of substitutes, such as benzol and alcohol, these products are now running to waste because of the lack of that need. Each year we are wasting—destroying—vast resources capable of producing engine fuel, because they are a little less convenient to utilize than the petroleum resource, although the latter is strictly limited in size. Only an inadequate policy would permit such sacrifice of ultimate value to the expediency of the moment, although such a procedure is of such common experience as to be looked upon as an economnc necessity, and hence justifiable. An analogy is afforded in the case of water-power, which is still largely unused, on the assumption that coal is plentiful and hence there is no need for its development. The blind adherence to this dictum, ignoring its inevitable bearing upon transportation, has probably already caused sufficient disaster to arouse suspicion of its wisdom.

A constructive economic policy, then, will not ignore benzol and alcohol, but on the contrary will promote their use. Benzol, indeed, demands such consideration on its own account, since a market for this product must be built up to help carry forward the important matter of proper utilization of coal. Alcohol, too, is not without claim on grounds outside the petroleum interest, for its fuel utilization would give an outlet to the increasing number of distilleries going out of service, while its possibilities as to generation on farms and its peculiar adaptability to tropical conditions form considerations of considerable weight. For the sake of the petroleum resource itself it would not be unwise to bring some relief to the demand for gasoline, which, if unaided, must face eventual curtailment.¶

SUMMARY

The petroleum resource stands out because of its limited size and decreasing availability, the growing importance of its products, and the notoriously high percentage of waste involved in its exploitation. According to conservative estimates, scarcely 10 per cent of the resource value is recovered under present conditions, while the unmined supply now available in the United States is only about 70 bbl. to the person. A survey of the resource and of the industrial activities engaged in its development indicates that the bottom cause of the present wasteful employment of this invaluable resource is a lack of adjustment between economic circumstances affecting production, and the unique geological conditions under which petroleum occurs. The geological unit or reservoir, by nature indivisible, is arbitrarily subdivided into small parts for purposes of individualistic production. This discordance leads to a train of wastes that consume the bulk of the resource. Its cause may be removed by reshaping the method of production so as to fit in with the occurrence of the resource, and the means for this accomplishment will come through the development and application of a constructive economic policy.

§We must remember that this country, thus far, has never had to face the exhaustion of a great resource. A somewhat analogous experience is afforded, however, in the case of the virgin forests.

||It is obvious that oil shale, to be profitable, must yield a full complement of products. There is still an oversupply of petroleum in respect to that consideration. An artificially stimulated or premature production of large quantities of shale oil would encourage the perpetuation of the current wasteful method of exploiting petroleum.

¶It may not be beyond the interest of the automotive industry to bend its energies toward providing a situation where benzol and alcohol will come into action. Such effort, if undertaken, should merit popular support because of its constructive tendency; and in particular will it stand in need of sympathetic governmental help when it approaches the legal aspects of alcohol exemptions.

GASOLINE ENGINES IN MINES

THE mining laws of some states forbid the use of gasoline or any petroleum product for generating power inside a mine. The laws of other states allow its use in engines, provided certain precautions are taken. In all mining states, however, there are many small or remote mines that must necessarily be neglected to some extent by mine inspectors.

In order to guard against fires underground, gasoline engines should not be installed in timbered places, and as little timber as possible should be used in the engine foundation. If a priming can is used it should be closed like an oil can and kept in a place where it will not be overturned or subject to damage.

Pure gasoline vapor entering a flame will burn quietly if surrounded with air, but if it is mixed with the proper proportion of air and then ignited it will explode violently if in the least confined. Under the conditions usual in mines, air containing 1.4 to 6.4 per cent of gasoline vapor will explode in the presence of an open flame. If any quantity of gasoline between ½ and 2 gal, be spilled or leak from a container and then vaporizes and mixes thoroughly with the air in a room 7 feet high by 9 feet wide by 15 feet long, it will form an explosive mixture throughout the room. Such conditions of thorough mixing are not common, the gasoline vapor being mixed with the air in different proportions at different points. Yet at any particular point the proportions may be such as to cause a violent explosion if the mixture be ignited.

Exhaust Gases

The exhaust from a gasoline engine is a mixture of gases, some harmless and some dangerous. The most dangerous is carbon monoxide. Carbon monoxide is the product of burning carbon without enough air. It is the most dangerous gas in what the coal miner knows as "after damp." One part of carbon monoxide in 1000 parts of air can be breathed for short infrequent intervals without discomfort or danger, but one part of this gas in five hundred of air may cause severe headache after it is breathed a short time. A little more carbon monoxide may cause unconsciousness and finally death. If the engine and its carbureter or mixing valve, are in perfect adjustment, little if any carbon monoxide will be given off. However, if all adjustments are not perfect, too much gasoline is burned and the exhaust contains considerable carbon monoxide. This condition is common in engines as used in mines. The presence of a dangerous amount of carbon monoxide in the exhaust or in the mine air, can be detected only by special tests. The gas is odorless and colorless and will not produce any effect on the flame of a safety lamp or candle unless it is present in such large amount as would cause almost immediate death to anyone breathing air containing it. Though air must be circulating around the engine to dilute the carbon monoxide to make it harmless. A gasoline engine can generate as much as 0.45 cu. ft. of carbon monoxide per horsepower per minute.

In most metal mines dependence is placed solely on natural ventilation. Changes in the direction of the wind, or in atmospheric conditions, may quickly reverse the direction in which the air is traveling. In planning for the disposal of the exhaust gases from gasoline engines safety can generally be guaranteed only by carrying the exhaust to the surface. If an airway in some part of the mine where men are never required to travel carries a strong return curent, it may be safe to discharge the exhaust therein. If that part of the mine in which the engine is working is ventilated by an exhaust fan, it would be safe to carry the engine exhaust into the return air-duct through which the fan draws the air. To turn the exhaust into the sump or into running water is useless, as carbon monoxide is practically insoluble in water. It will bubble up through water without being changed in any way except that the smoke and the odor of accompanying gases may be removed, the poisonous quality of the gas remaining unchanged. Thus the possible warning of the presence of carbon monoxide by the odor of accompanying gases will be removed if the exhaust is turned under water.

Handling Gasoline

In drawing gasoline or pouring it from one container into another static electricity may be generated. It has been shown by experiment that unless the containers are grounded and are in electrical connection, the accumulated electricity may discharge by a spark sufficiently strong to ignite the gasoline vapors. The use of portable tanks will entirely eliminate this danger within the mine. At the filling station the tanks should be connected electrically in order to prevent such accumulations. Cans with handles of wood or any other insulating material should never be used. If a hose with a metal nozzle is used, a bare copper wire inside the hose and soldered to the nozzle and to the tank connection will complete the circuit. If a can and a funnel are used, they may be attached to the tank with light copper or brass chain.

Many engines are built with cavities or inclosed spaces in the crank-case, base or some other part. These may be full of gasoline vapors, and when inspecting or making repairs with an open light, men have been severely burned when these vapors have ignited. To guard against such accidents all cavities should be blown out with compressed air or steam. If neither of these is available the cover should be removed, the vapors fanned out, and a lighted lamp or candle attached to a long stick passed around inside the cavities to burn out any that may remain, before the workman puts his hands or face near them.

Unless an ample current of air at considerable velocity is passing, gasoline should not be used to clean an engine or other machinery. Even if there be sufficient air to sweep away the vapors as soon as they are given off, open lamps should always be kept at a safe distance and on the intake side, so that the vapor can not be carried to the light.

Burning gasoline floats on water and thus spreads the blaze. The best method in case of fire is to use a reliable chemical fire extinguisher. If none is at hand, the fire should be smothered with sand or sawdust or heavy cloths.

If precautions are taken and due care is exercised in using gasoline, its use as engine fuel offers little, if any, more danger than that of other forms of mechanical power.

—Kudlich and Higgins (Bureau of Mines)

STANDARDIZATION

APART from the special conditions induced by the war, a standard or specification should not be an end in itself and should not be based on average production. It should be an aim, an ideal for the manufacturer to work up to, rather than a schedule representing the medium of existing productive capacity. On first thought it would appear to be to the interest of the manufacturer to place a standard as low as possible to ease the process of making by the production of an inferior article. This is an old fallacy, which should have been exploded long since. By raising the standard to the best possible attainable, by tightening up the specifications, a manufacturer is compelled to give us of his best, and where an entire industry is so compelled, the result will certainly in the end contribute to the prosperity of the Nation.

—*Aeronautics.*

The High-Compression Oil Engine

By W. G. Gernandt* (*Member*)

Mid-West Section Paper *Illustrated by* Charts

THE necessity of providing suitable means to burn the heavier oils in internal-combustion engines successfully is slowly but surely being brought home to designers and builders of automotive apparatus, and the time seems opportune for making a comparison of the various methods of injecting liquid fuel into the combustion chamber of an engine, with a view to impressing upon the engineers the need of using the high-compression type of engine for the proper burning of the heavier oils.

To really show the advantage of high compression, it will be necessary to describe in detail in a somewhat semi-technical way the cycle of events taking place in the Otto, or constant-volume engine, as well as the Diesel, or practically constant-pressure engine.

Four-Stroke Constant Volume or Otto Engine

The Inspiration Line AB At the end of the exhaust stroke, *E-F-A*, Fig. 1, the clearance volume V_c is filled with burned gases under a pressure of P_e. When the piston moves on its inspiration stroke the pressure falls from P_e to the inspiration pressure P_i along a curve determined by the reexpansion of the products of combustion remaining in the combustion chamber at the time the exhaust valve closed. Only after the reexpansion will the fresh charge be drawn into the cylinder. It is thus seen that the volumetric efficiency E_v of the cylinder, which equals the ratio of volume of fresh charge to the volume of piston displacement, depends directly upon the volume and pressure of the burned gases remaining in the cylinder or combustion chamber. If through a poor design of combustion chamber, too small an exhaust valve, faulty valve-timing or a restricted exhaust manifold, the exhaust pressure should remain too high, the effect of reexpansion before actual inspiration will be greatly increased. However, with a properly designed engine this loss in volumetric efficiency will be low and depend directly upon the actual volume of the combustion chamber.

Another factor affecting this efficiency is the inspiration pressure P_i. At the end of the inspiration stroke the cylinder contains a volume of gas V_t made up of products of combustion and a fresh charge under a pressure of P_i. The compression stroke raises the pressure of this gas to P_c at a volume of V_c. The compression curve *BC* crosses the atmospheric line *XX* at *G* when the stroke volume is only V_{si} and not the full volume V_s. Hence, the volume $V_s - V_{si}$ represents a loss in volumetric efficiency which is greater as P_i becomes smaller. Therefore, in order to keep this efficiency high, inlet pipes and valves should be of such size as to reduce the inspiration friction to a minimum, thereby keeping the inspiration pressure as close to the atmospheric line as possible. A volumetric efficiency in excess of 100 per cent could be obtained if it were possible to force the charge into the cylinder at a greater pressure than atmosphere. It is well to note also in this regard how the carbureter reduces the volumetric efficiency: *a*—To atomize the fuel properly it is necessary to choke the air at the spray nozzle so as to increase the gas velocity at this point. This causes an increased friction, thereby decreasing the value of E_v. *b*—With the present low grade of fuel it is necessary to add heat to the intake air to increase the atomization of the fuel. This is done by drawing the air supply from a jacket surrounding the exhaust pipe. Naturally the value of E_v is reduced because of the reduced weight of the charge and the increased friction in inlet pipe. *c*—At reduced speeds and powers the throttle-valve of the carbureter is partly closed, causing increased friction and a reduction of the inspiration pressure P_i. Thus, it is seen that the value of E_v is reduced by an increase of compression space, the use of a carbureter and the necessity of using heat to atomize properly the poor grade of fuel now on hand.

The Compression Line BC The compression line may be taken to follow the general law of $pv^n =$ constant. In other words, the pressure is increased in a certain proportion to the decrease in volume, depending upon the kind of gas compressed, the temperature of the gas at the beginning of compression and the amount of heat given to or taken from the cylinder walls. For air the value of n is 1.41, but for a mixture of fuel and air it varies from 1.30 to 1.38, with a good average for actual conditions of 1.35. As the thermal efficiency of an engine increases slightly with an increase in the value of n, it is seen that in compressing pure air a decided advantage would be gained.

It has been proved that the thermal efficiency of an engine depends upon the pressure at the end of compression. The higher this pressure the higher the efficiency and, consequently, the greater the fuel economy. In a constant-volume, or Otto engine, the explosive mixture is compressed in the combustion chamber; therefore, the limit of compression pressure is the preignition point of the mixture. For gasoline the compression pressure can be safely carried to 100 lb. per sq. in., while for kerosene the pressure must be reduced to about 65 lb., showing not only a loss in thermal efficiency, but a loss in power output as well when using kerosene for fuel in this type of engine.

The Combustion Line CD The shape of this line depends upon the composition of the fuel mixture, the point of ignition and the piston speed, presupposing a properly designed combustion chamber. The fuel mixture composition depends upon the kind of fuel used, and the proper commingling of the fuel and air. This commingling is the function of the carbureter and upon its proper adjustment depends the explosibility of the charge. A too rich or too weak mixture causes the burning to lag and the maximum pressure obtainable from the fuel will be greatly reduced. The point of ignition plays an important part in the shape of the combustion line in that should the ignition be late, not only is the maximum pressure in the combustion chamber reduced but instead

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of the combustion line being vertical it will fall away as shown in the dotted line in Fig. 1. If the ignition be too early, the pressure rise will occur before the piston has reached its dead center on the compression stroke, and a decided loss in power and economy and a great strain on the crank bearings will result. It is therefore essential to have ignition occur at the proper point. Piston speed also affects the combustion line in that with an increase in speed the combustion must take place at a greater rate and therefore the ignition point must be advanced to maintain maximum power and economy conditions.

The Expansion Line DE The expansion line, like the compression line, follows the general law of pv^n = constant, but the value of n is approximately 1.38 be-

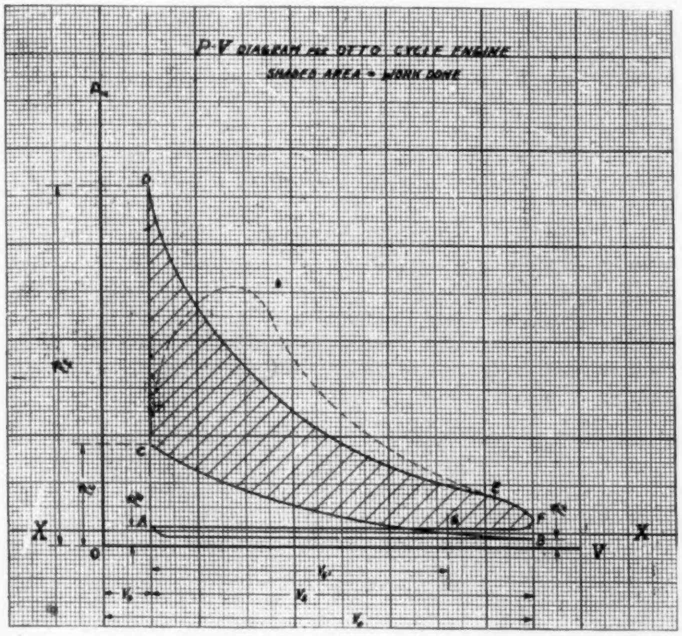

FIG. 1—P-V DIAGRAM FOR OTTO CYCLE ENGINE
THE SHADED AREA INDICATES WORK DONE

cause the burned gas resembles air more closely than does the mixture during compression. As in the combustion line, the quality of mixture, time of ignition and piston speed have a great bearing upon the expansion line. However, other conditions, such as cooling and length of stroke also play a part in the power output and fuel economy, but with proper design they are of less importance.

The Exhaust Line EFA This line is dependent upon the size of the exhaust ports and the exhaust valve-timing. It is absolutely essential to allow for the most rapid drop in exhaust pressure to atmospheric to insure the lowest possible value for P_e, thereby increasing the volumetric efficiency, as shown under the inspiration line.

The requirements for best efficiency of combustion and expansion have been stated briefly, but in greater detail they are as follows:

A Highest possible compression before ignition. The effect of this is
- Reduced combustion chamber with a reduced admixture of products of combustion
- 2.2 Less heat loss to water-jacket because of reduced combustion chamber
- Greater mean effective pressure for the power stroke
- Lower terminal or exhaust pressure and exhaust loss
- Greater ease of ignition of the charge

B Pure uniform mixture and rapid combustion to avoid after-burning with a consequent loss to the water-jacket

C Proper piston speed and bore-stroke ratio to keep the jacket loss as low as possible

D Reduction of inlet and exhaust friction to the utmost

E Proper valve-timing

THE DIESEL OR HIGH-COMPRESSION ENGINE

The Inspiration Line AB As in the constant-volume cycle there remain in the combustion chamber products of combustion at a pressure of P_e, and this gas must be reexpanded before inspiration can take place. However, it will be noticed that this reexpansion takes place at a more rapid rate, as the volume of combustion chamber is greatly reduced because of the greater compression required for this type of combustion.

The next point to be considered is the inspiration pressure P_i. In this case the same conditions govern the value of P_i as in the constant-volume engine with the exception that pure air is drawn into the cylinder instead of a mixture of fuel and air formed by the carbureter. It is instantly seen that in eliminating the carbureter the suction friction is reduced, the temperature of the air is not increased and the throttle friction at decreased speeds and powers is entirely eliminated because engine control is produced by varying the amount of fuel supplied during combustion. Therefore, pre-

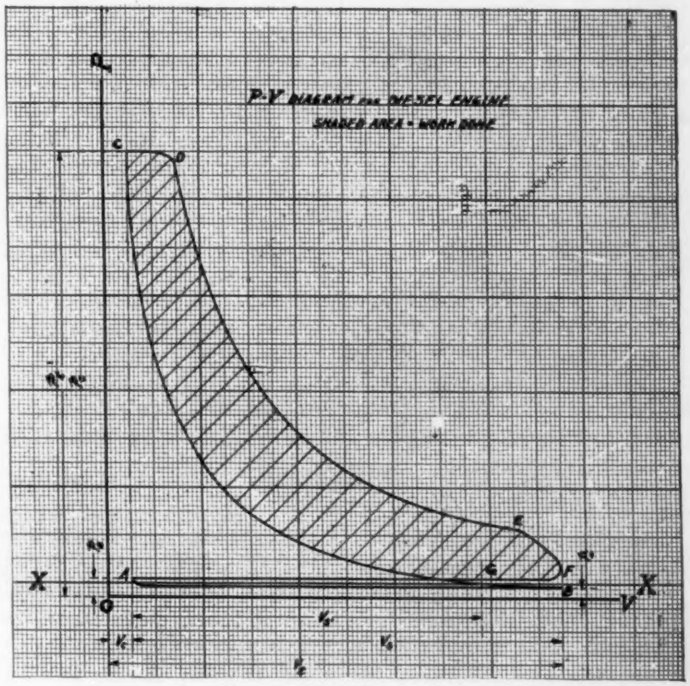

FIG. 2—P-V DIAGRAM FOR DIESEL ENGINE
THE WORK DONE IS INDICATED BY THE SHADED AREA

supposing equal-sized valves in both cases, the value of P_i will more nearly equal atmospheric pressure in the case of the constant-pressure engine, as shown in Fig. 4.

The Compression Line BC The compression line follows the same general law as in the constant-volume engine (pv^n = constant), but the value of n reaches 1.41, which allows for a slightly increased thermal efficiency.

As pure air is compressed, preignition is not the limiting factor in determining the compression pressures, and pressures as high as 600 lb. per sq. in. can be used. However, as this great pressure causes an increase in the weight of the various parts out of proportion to the gain in thermal efficiency, 500 lb. is considered a good

limit of compression for this type of engine. Thus the main point in increased thermal efficiency has been gained without restriction as to fuel used or preignition.

The Combustion Line CD Aside from the high-compression feature, the combustion line differs considerably from the constant-volume engine and requires further attention. Instead of igniting a mixture of fuel and air electrically with an attendant sudden rise in pressure, the fuel is injected into an atmosphere of highly compressed and heated air when the piston reaches the upper dead-center, causing the fuel to be burned in suspension, or as it enters the air. This method of burning causes a slight rise or reduction in pressure during combustion, depending upon the rate of injection and the thoroughness of the atomization. While it is not possible to increase the combustion pressure in as great a ratio as in constant-volume combustion, considerable increase can be obtained depending upon the method of injecting the fuel. If the injection is relatively slow and takes place over a considerable portion of the power stroke, the pressure will drop. As the injection rate is increased the pressure drop decreases until a rate is reached where a decided increase in pressure during combustion is noted. The various methods used to inject fuel into the combustion chamber will be described in detail later.

The Expansion Line DE As in the case of the constant-volume engine, the expansion line follows the general law pv^n = constant, with the value of n equaling approximately 1.38 as in the previous case. However, due to the high pressure attained during compression, the rate of expansion is greater in this case and therefore the pressure in the cylinder at the time the exhaust valve opens is lower. The effect of this is to reduce the temperature and consequently the heat loss during exhaust. This means an increase in thermal efficiency.

The Exhaust Line EFA This line is similar to the exhaust line in a constant-volume engine, but as the pressure and temperature at the time of exhaust-valve opening is lower, there is a slight difference in the value of P_e. However, in actual practice the difference is so slight that it is very difficult to measure its value.

From the foregoing it is seen that the high-compression type of internal-combustion engine more nearly approaches the requirements for best efficiency enumerated before than does the constant-volume engine.

Comparative Diagrams

A comparison of pressure-volume diagrams computed for a 4 by 6-in. engine, Fig. 3, will bring out more clearly the actual differences in the two types of engine. In computing these diagrams the following assumptions were made to make comparison under identical conditions:

(a) Compression pressures—Otto, 108 lb.; Diesel, 500 lb.
(b) Ignition pressure—Otto, 400 lb.; Diesel, 500 lb.
(c) Value of n, 1.35 for both compression curves
(d) Value of n, 1.38 for both expansion curves
(e) That ignition took place instantaneously in the case of the Otto engine and that ignition took place for 15 deg. of crank travel in the Diesel engine, expansion beginning immediately at end of combustion
(f) That the exhaust valves opened when piston was ½ in. from the lower dead-center

Aside from the difference in shape of the two diagrams it is well to note the actual difference in total cylinder volume between the two types of engine. The distance from the ignition line to the line marked "clearance line" is the actual volume of burned gas remaining in the combustion chamber, and it is this volume of gas that must be reexpanded before inspiration of the charge can take place. Thus it is instantly seen that the volumetric efficiency which depends directly upon this volume is greater in the case of the Diesel engine. The actual result of this can be seen in the upper portion of Fig. 3. Also, because of this difference in volume, there is less heat loss to the cooling water from the combustion chamber in the Diesel than in the Otto engine.

In the case illustrated the length of the diagram is 6 in., the pressure scale is 100 lb. per in. in height, the area of the Otto diagram is 5.15 sq. in., and the area of the Diesel diagram is 4.94 sq. in. This gives 86 lb. per sq. in. as the mean effective pressure for the Otto and 82 lb. per sq. in. for the Diesel engine, showing that more power is actually developed by the Otto than by the Diesel engine per square inch of piston area. However, as the efficiency of the Diesel is superior to that of the Otto engine, it will develop more horsepower per pound of fuel burned. Again, the rate and method of fuel injection can be altered so as to cause the combustion pressure to be increased considerably over the compression pressure, in which case the area of the diagram and consequently the horsepower developed would be increased beyond that shown by the Otto engine. Thus, we can expect, with proper fuel injection, equal or greater powers compared with the Otto engine of the same bore and stroke.

Another point well worth mentioning again is the difference in pressure at the time the exhaust-valve opens. In the case illustrated the difference is actually 10 lb. This means that more heat has been abstracted from the fuel for actual power purposes and less heat given to the exhaust, with a net result of increased thermal efficiency.

Diesel Method of Injection

In the Diesel engine the fuel is injected into the combustion chamber by compressed air at a pressure ranging from 700 to 1000 lb. per sq. in. To accomplish this it is necessary to provide a fuel pump, a two or three-stage air compressor, a compressed-air tank and an air-injection tank or bottle. The fuel pump is designed so that it delivers a measured quantity of fuel to the injection valve, the actual amount of fuel being regulated by a governor according to the load on the engine. As the fuel-injection valve is always under the injection air-pressure, it is necessary for this pressure to be overcome before any fuel reaches the valve. Therefore, the fuel pump must be capable of maintaining a pressure of 800 to 1200 lb. per sq. in. upon the fuel during the delivery period. The air-compressor is so designed that cold air at a pressure of 700 to 1000 lb. is delivered to the injection-air tank. This necessitates the use of a cooler between each stage of compression, and one between the last stage and the tank. The use of the tank is for starting purposes.

As the inspiration and compression strokes are common to all types of engines and the method of injection is the main feature under discussion, a detailed description of the injection and combustion period will be of great interest. During the inspiration stroke, a measured quantity of fuel is delivered near the bottom of the fuel injector just above the needle-valve or spray-nozzle. When the needle-valve is opened, the injection air, which in all cases is well above the compression pressure, is forced against the fuel, atomizes it and forces it into the cylinder. One point worthy of note is the fact that the action of the injection air thoroughly atomizes the fuel prior to injection, and upon this atomization depends the combustion efficiency of the cycle. The amount of fuel, as before stated, is regulated by the fuel pump and

governor. The duration of injection or combustion is also regulated according to the load on the engine. For light loads the injection valve remains open a shorter period than for heavy loads. Aside from the advantage of thorough atomization there is one great objection to this method of injection, namely, refrigeration during injection. As the compression pressure is about 500 lb. and the injection pressure between 700 and 1000 lb., there will be a very rapid expansion of the injection air from the maximum to the compression pressure. This rapid expansion causes a great reduction in the temperature of the injection air and fuel, just as rapid compression causes an increase in the temperature. The effect

Summarizing the advantages and disadvantages of the Diesel method of injection, they are as follows:

ADVANTAGES

(a) Fuel injection mechanically timed
(b) Fuel thoroughly atomized prior to injection
(c) Widely different fuels can be used without alteration
(d) Rate of burning controlled mechanically
(e) Two-stroke principle can be used successfully

DISADVANTAGES

(a) Refrigeration during injection
(b) Reduced flexibility
(c) Use of a high-pressure fuel pump
(d) Use of an air-compressor, coolers and tanks
(e) Troublesome to start

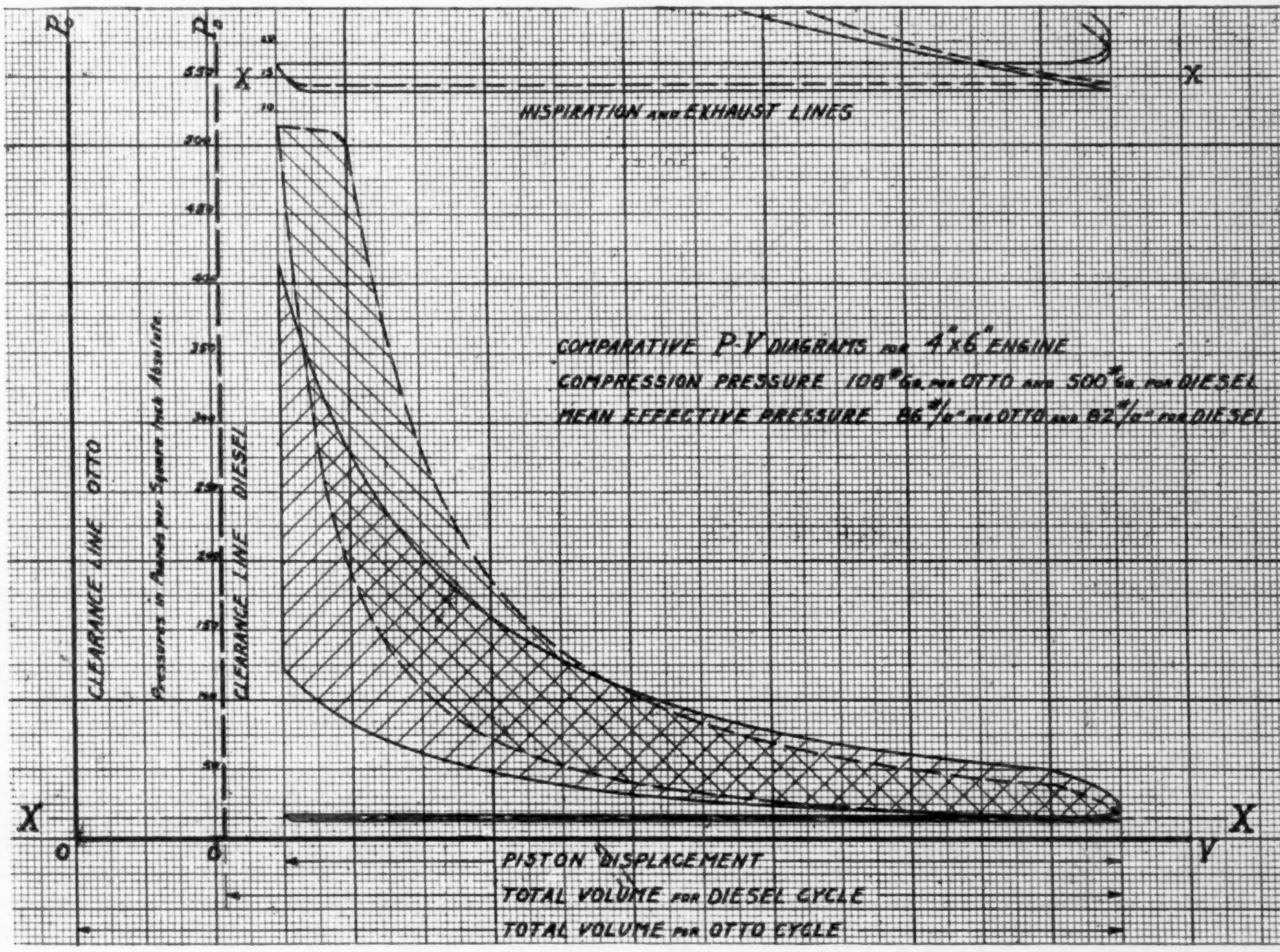

FIG. 3—COMPARATIVE P-V DIAGRAMS FOR A 4 X 6-IN. ENGINE OPERATING ON THE OTTO CYCLE AND THE DIESEL PRINCIPLE
THE RELATION OF THE INSPIRATION PRESSURES TO EACH OTHER AND TO ATMOSPHERIC PRESSURE IS BROUGHT OUT IN FIG. 4 AT THE TOP OF THE DIAGRAM

of this sudden reduction of temperature causes a time lag in the combustion—because cold fuel is difficult to ignite—as well as the collection of small particles of fuel on the piston, at which point local burning occurs. The result of this is an invariable sag of the piston head due to excessive local temperature. Were it possible to use highly heated air for injection, this refrigeration would to a great extent be overcome. However, with the Diesel method of injection, combustion would take place at the injection valve prior to the injection period. It is well to note that in the Diesel engine the fuel injection is in mechanically timed relation to the piston position, but as the injection and compression pressures are always constant, the rate of injection is also constant, which limits the engine's flexibility.

Hvid Method of Injection

In the Hvid engine the fuel is injected into the combustion chamber by a pressure rise in the fuel cup, caused by the explosion of part of the fuel charge, and the operation takes place as follows: During the inspiration stroke of the piston, fuel and air are drawn into the fuel cup through mechanically timed valves, simultaneously with the filling of the cylinder with pure air through the main inlet-valve. On the compression stroke the fuel and air in the cup become heated to such a point that combustion of the lighter or more volatile particles of the fuel takes place. This causes a sudden rise in the pressure within the fuel cup above that in the main combustion chamber, with the result that the fuel is atomized and forced into the highly heated air in the

cylinder. The fuel is burned in suspension or directly as it enters the heated air, causing a slight rise in pressure within the cylinder. From this point on the expansion and exhaust strokes are identical with those of any internal-combustion engine.

As in the Diesel engine, the amount of fuel admitted to the cylinder is governed by the load on the engine, but the ignition is not mechanically timed. The control of the time of ignition is entirely by use of proper sized injection holes in the fuel cup for a given compression pressure and fuel. Although this method of timing the ignition proves quite satisfactory over a considerable range of speeds when the fuel and compression remain constant, should there be any variation in the compression pressure or the fuel used, a considerable variation in the time of injection would occur.

It would appear from this that a multiple-cylinder engine would not operate as successfully as a single-cylinder engine, unless the compression pressure were uniform in all cylinders. With this method of fuel injection it is not possible to use the two-stroke cycle principle, because the charging of the fuel cup with air is absolutely necessary to form an explosible mixture. This charging is done during the suction stroke of the piston and would be very difficult to accomplish if inspiration and exhaust occurred simultaneously, as in the two-stroke engine. Thus it is seen that the Hvid principle is limited to the four-stroke principle of operation and, therefore, is limited relative to the weight per horsepower.

Summarizing the advantages and disadvantages of the Hvid method of injection, they are as follows:

ADVANTAGES

(a) No working parts for the injection of the fuel
(b) Fuel thoroughly atomized prior to injection
(c) Fuel heated prior to injection
(d) Flexibility much greater than Diesel
(e) Easily started
(f) Fuel economy greater than Diesel because of heating fuel

DISADVANTAGES

(a) Fuel injection not mechanically timed
(b) Inability to use the two-stroke principle of operation
(c) Necessity of changing fuel cup for widely different fuels

McClintock Method of Injection

In the McClintock engine the fuel is injected into the combustion chamber by compressed air trapped in a separate chamber during compression. The operation of this engine is as follows: When the pressure in the cylinder is the least, or during inspiration, fuel, which is under a slight pressure maintained on the fuel tank by a small air pump, is deposited in a small chamber in direct communication with the combustion chamber through small injection tubes. During the compression stroke the air is compressed into a separate chamber through an automatic valve. As the clearance between the head and the piston is only that necessary for mechanical purposes, practically all of the air in the cylinder is compressed into this chamber.

When the piston starts on its power stroke, the air from the chamber is by-passed into the combustion chamber through a mechanically timed valve and a Venturi-shaped port into which project the fuel tubes. Because of the shape of the air port, the fuel is drawn from a small chamber and an intimate mixture of air and fuel takes place, with the result that the combustion begins immediately and with a close resemblance to the Bunsen burner. The amount of fuel as well as the duration of combustion is controlled mechanically by the load on the engine. With this method of burning, extremely high temperatures result, and the materials for the injector and Venturi tubes must be very carefully selected.

From a standpoint of combustion, this engine possesses the advantage of an intimate mixture between the fuel and the air needed for combustion. However, as the piston must move on its down stroke a considerable distance before the transfer of air takes place, combustion comes late in the stroke, with a resultant high exhaust pressure and temperature. The engine is, by the way, just as flexible as most automobile engines.

Summarizing the advantages and disadvantages of the McClintock engine, they are as follows:

ADVANTAGES

(a) Fuel thoroughly atomized and mixed with the air
(b) Fuel heated prior to injection
(c) Flexibility extremely good
(d) Rate of burning can be mechanically controlled
(e) Two-stroke principle can be used successfully
(f) Fuel economy higher than Diesel
(g) Widely different fuels may be used without alteration

DISADVANTAGES

(a) Complicated valve mechanism
(b) Fuel injection is not mechanically timed
(c) Somewhat troublesome to start—similar to Diesel
(d) Advantage of high compression partially lost prior to combustion

Gernandt Method of Injection

In the Gernandt engine the fuel is injected into the combustion chamber by super-compressing a portion of the products of combustion which have been trapped at the time the pressure in the cylinder has attained its maximum. Mechanically, this can be accomplished in various ways, depending upon the general design of the engine, and the trapping chamber may be actually sealed by the use of valves between the super-compressing means and the combustion chamber, or be in direct communication with the combustion chamber through the very small injection holes. In either case the injection method is identical and takes place as follows:

During the suction stroke in a four-cycle, or during simultaneous exhaust and inspiration in a two-cycle engine, fuel is deposited in a small chamber between the combustion chamber and the super-compressing means, either by gravity or under a slight pressure maintained on the fuel in the tank. The fuel is metered and passes through a mechanically timed valve. During the compression stroke the fuel attains temperature and the pressure rises in the fuel chamber. When the piston reaches its upper dead-center, the products of combustion, previously trapped, are super-compressed mechanically and forced through the fuel chamber and into the combustion chamber.

Thorough atomization takes place during the injection period, as in the Diesel engine, but in this case the injection gas is highly heated and refrigeration has been practically eliminated, the amount of the products of combustion necessary for injection being so small that the burning effect has not been impaired. Also, there is no burning of the fuel until it is actually injected into the combustion chamber. In this engine the fuel, the injection and the rate of injection are mechanically timed, and with an increase in the crankshaft speed there will be a corresponding increase in the rate of fuel injection because the fuel must enter the combustion chamber during a certain angular travel of the crankshaft and

not during a certain fixed time. Thus the flexibility of this engine will be very good.

Actual tests on this method of injection have shown promising results, and the speed limitation of a multiple-cylinder engine will depend only upon the combustion rate of the fuel, which, in turn, is dependent upon the thoroughness of atomization, the rate of injection and the shape of the combustion chamber. Experiments are now under way to improve the fuel combustion rate.

Summarizing the advantages and disadvantages in this method of injection, they are as follows:

ADVANTAGES

(a) Fuel injection mechanically timed
(b) Fuel heated prior to injection
(c) Fuel thoroughly atomized prior to and during injection
(d) Rate of burning mechanically controlled
(e) Widely different fuels can be burned without alteration
(f) Fuel economy is greater than Diesel because of preheating
(g) Greater flexibility
(h) Two-cycle principle can be used
(i) Engine very easily started

DISADVANTAGES

(a) Addition of super-compression means, making engine more complicated than the Hvid but much less so than the Diesel

In conclusion, the main point to be brought out is that regardless of the method used for the injection of fuel, advantage should be taken of the high economy and the possibility of burning the heavy fuel oils actually accomplished by the use of the high-compression oil engine.

Discussion

QUESTION:—I would like to ask Mr. Gernandt if the principle he mentions is his own idea, and if it could be built into a small, a medium or a high-speed engine.

ANSWER:—Yes, I can answer to both questions. The engine we first used was a 3 by 5-in. single-cylinder. The results we got from this were nothing to boast of, but we built a second engine, a 4 by 6-in. single-cylinder, and it runs very smoothly. It is a stationary engine. The speed ranges from about 200 to 860 r.p.m. and the reciprocating parts weigh from 16 to 18 lb.

QUESTION:—Would there be anything to interfere with its use on a multiple-cylinder engine?

ANSWER:—No, there would not.

QUESTION:—Could the speed limit be increased if the atomization of the fuel was improved?

ANSWER:—The speed limit depends upon the combustion rate of the fuel. The higher the atomization of the fuel, the better and quicker is the burning. In this way the combustion rate is increased.

QUESTION:—Could it be further increased by having more points of injection?

ANSWER:—The number of injection holes has a great deal to do with it, as the combustion rate depends upon the proper distribution of the fuel. These holes are about No. 56 drill, and there are three. They are so spaced as to permit the quickest union with the air.

QUESTION:—What is the heaviest fuel that has been tried in this type of engine? Has Pennsylvania or asphalt base been used?

ANSWER:—We have not gone into that very deeply. Experimenting is now going on. As a matter of fact we operated on lubricating oil taken from the crankcase of an automobile. It was pretty black. Possibly it was only half fuel. I do not think there is any limit to the heavy fuel one can burn in this engine.

QUESTION:—Could you use this method on a 4 by 6-in. engine, running 1000 ft. per min. piston speed?

ANSWER:—Yes, we are using it on an engine of that size now.

QUESTION:—Is there greater flexibility in the two-cycle than in the four-cycle engine, and a greater amount of power per pound of engine weight?

ANSWER:—Yes, there is. The greater the impulses per cylinder or impulses per minute, the greater the flexibility. Flexibility depends upon the number of impulses as well as upon the weight of the reciprocating parts.

QUESTION:—Would there be any difficulty in scavenging a two-cycle engine?

ANSWER:—No, because in this type of engine the scavenging agent is pure air, not a mixture of fuel and air. In the ordinary type of two-cycle engine using a carbureter the ports must be designed so as to scavenge the exhaust gas perfectly and not lose any of the scavenging mixture. You can lose a lot of air through scavenging and yet not lose much in power.

QUESTION:—Have you made tests to find at what particular position the injection should take place to get the best fuel economy?

ANSWER:—No, we have not.

QUESTION:—The combustion period is short. Is there not some particular way of injecting the fuel to get the highest efficiency?

ANSWER:—Yes, there is. As I have stated, experiments are being conducted to ascertain the best position for the injection holes, their correct size and the proper rate of injection. In injecting heavy oil the fuel must get to the air, and it has to be done rapidly, the quicker the better. It can be done, however, by a correctly shaped combustion chamber, injecting holes of the proper size and a maximum rate of plunger travel.

QUESTION:—Do you need the same compression pressure as the Diesel engine?

ANSWER:—Yes; about 450 to 500 lb., depending upon the fuel used.

QUESTION:—What is the relative size of your plunger?

ANSWER:—Three-quarters inch diameter by 1/4 in. stroke.

QUESTION:—Is the plunger cam operated?

ANSWER:—Yes. The injection takes place in about 35 deg. of crank travel. In the Diesel engine the time required to inject fuel is around 45 to 50 deg. Injection occurs about 8 deg. before dead-center and continues from 38 to 42 deg. of crank travel past dead-center.

QUESTION:—You regulate the power by regulating the fuel, but such a small amount of fuel is used per stroke that you must have difficulty in controlling your engine to very low speeds.

ANSWER:—The control of the engine is entirely by the amount of fuel burned. The needle-valve regulates the quantity of fuel. We have run as low as 200 r.p.m.

QUESTION:—At high speeds the time limit is short, and the fuel holes are so small that the super-compression space might not be totally filled. Will there be pressure enough for fuel injection?

ANSWER:—Super-compression goes far beyond the compression pressure so that there is always an excess of pressure no matter what the speed of the engine may be. The plunger begins its downward stroke a short time before upper dead-center and continues about 30 deg.

past dead-center. In larger engines the super-compression chamber may be sealed with a valve between the combustion chamber and the super-compression means and actually trap the high-pressure gas.

QUESTION:—You claim that the efficiency of the Hvid engine is above that of the Diesel. Will you explain how that can be?

ANSWER:—It is due to the heating of the fuel in the Hvid engine. There is a very rapid explosion in the fuel cup that thoroughly atomizes the fuel. In the Diesel engine the injection air is cooled and they aim to put the fuel into the combustion chamber just ahead of the injection air to reduce the refrigeration action as much as possible. Efficiency really depends upon the degree of atomization.

QUESTION:—In the Diesel there is the onrush of the heavy pressure air whereas the Hvid has only the small injection holes for atomization.

ANSWER:—The Diesel has the same thing. The Diesel mixes air and fuel before the injection and atomizes the fuel during injection through small holes, as in the Hvid.

QUESTION:—Do you think the Hvid engine draws nothing but pure air into the cylinder during the suction stroke? The fuel goes in, and comes out of the holes in the center.

ANSWER:—Some of it does, but the amount is very hard to determine. There is some burnt gas, the same as in the Diesel. Their fuel goes in on the opposite side of the injection holes, and they depend upon the time element and the distance of travel being such that the fuel actually remains in the cup.

QUESTION:—Is there a knock in the Hvid engine?

ANSWER:—I believe there is. The problem with them is the timing of the fuel injection. It is done entirely by the size of the holes in the fuel cup. For certain compressions they have holes of a certain size. If the holes are correctly planned for a compression of say 450 lb. and there should be a drop in compression, the holes would be too small and the fuel would go into the combustion chamber late. If the compression were increased above 450 lb., the holes would be too large and fuel injection too early. It is very difficult to design for variations in compression. It also is difficult to keep the compression balanced in a multiple-cylinder engine, and it is upon this compression balance that the Hvid principle of injection depends. With a hole too large the fuel would enter the cylinder early and it is possible to get a knock. The same thing would happen if a lighter fuel were used. As to the relative atomization of the Diesel and the Hvid engines, the Diesel injection valves are designed so as to present a labyrinth passage for the air and fuel. This insures thorough atomization and better burning in place of refrigeration effects. I do not know that it is due to high burning. It is more of an injection of highly atomized vapor than two or three spurts of flame. You get partially the same effect in breaking up the fuel by an explosion, as in the Hvid, but the injection is that of two or three spurts of flame.

SAWDUST A SOURCE OF SUBSTITUTE FUEL

BENEFIT to automotive industries, the elimination of a nuisance and the release of food products are claimed in an article on a new source of alcohol fuel appearing in a recent issue of *The Motor* (London). The plan proposed contemplates the use of sulphuric acid produced on a scale which while only adequate to meet the demands of war, greatly exceeds the normal peace requirements, and sawdust and chipwood which are now going to waste. At the present time the uses of the sawdust are making roads, as a fertilizer and as a source of pulp for paper making.

The conversion of sawdust into alcohol was first commercially agitated in the United States about 1909, and it is stated that it is possible to produce absolutely pure alcohol from sawdust for about 10c. per gal. including the cost of barreling. To produce this quantity of alcohol 6 cu. ft. of sawdust, which costs about 26c. per ton, are required, and the actual cost is approximately 4c. per gal.

SAWDUST DISTILLATION

In producing alcohol in this way the sawdust is turned into a cylindrical vessel known as a digester and is acted upon by a mixture of steam and sulphuric acid which is sprayed into the vessel as it revolves. The acid acts upon the starch contained in the wood fibers, while the heat volatilizes the resinous contents, including the turpentine. After the digestive action has been completed the steam and resinous gases are exhausted from the digester under water, and in this way the acid is absorbed and the turpentine becomes liquefied and is thus readily recoverable. The material is then removed and deposited upon the screens of what are known as the diffusion batteries. Here it is washed by pure water, the cells through which the water is passed being connected in series. As the water flows through the cells it becomes more and more enriched with the sugar solution and the next steps are the removal of all traces of acid and the neutralizing of the sugar solution. As the presence of the acid prevents fermentation which is essential to the process, its removal is imperative. The neutralization is accomplished by the addition of lime and the liquor is then pumped into a tank and left to settle. The solid matter precipitates out and forms a sludge at the bottom of the vessel, while the clear liquid on top is pumped to the fermenting vat where yeast is added, about ½ bu. being sufficient to start the contents of a 3000-gal. tank.

The fermentation occupies about three days, and the resultant product is about 75 to 80 per cent alcohol derived from approximately 6000 lb. of dry wood waste. If the spirit is refined to give a 95 per cent product a yield of about 62 gal. is obtained, which is equivalent to about 21 gal. of high-grade alcohol per ton of dry wood. The product is said to be a pure white potable spirit free from any traces of wood flavor and superior to alcohol normally produced from grian. The process after the fermentation is started is the same as in the production of grain alcohol.

Denatured alcohol such as would give satisfactory results in an internal-combustion engine can be made at a lower figure than that required for the production of commercially absolute alcohol. It is stated at an American factory that 100 tons of wood waste will produce from 1200 to 1500 gal. of alcohol per day.

RESULTS IN FRANCE

In France it is possible to secure an even higher yield, about 20 gal. of alcohol being derived from 1 ton of wood waste. This is due to carrying out the digestive process for a further length of time, thus giving the sulphuric acid ample time to attack the wood thoroughly. In this process not only is the sulphuric acid recovered but acetic acid and stock food are obtained. In an experiment carried out shortly before the war 3200 lb. of wood waste were found to yield 21½ gal. of 94 per cent alcohol, 76 lb. of acetic acid and 1800 lb. of stock food. The last was compressed into briquettes and mixed with magnesia, the result being an excellent grade of artificial stone.

The Principles of the Wheeled Farm Tractor

By Edward R. Hewitt* (*Member*)

Annual Meeting Paper *Illustrated with* Charts

The hydrocarbon engine has in recent years placed us in possession of a light source of power for the purposes of farm traction. It is easily controlled and operated by one man. This has given the farm tractor problem quite a different position from that which it occupied when steam was the only method of propulsion. Although it was possible to make a one-man steam machine, this was not usually done, and the steam plant always had so many disadvantages in starting, use of water, leaking connections, etc., that its extended use was not to be expected.

Given a light source of power occupying comparatively small space, a tractor can take almost any form and yet do some work. While there is certainly vast latitude in design, and many different forms accomplish the purpose equally well, the fundamental principles governing the problem lay down certain postulates which must be followed if we are to reach any reasonable degree of engineering efficiency. It is to the study of these principles that I wish to draw attention, as their neglect has caused tremendous financial losses to both the manufacturers and the public. I will take up the problems, beginning, as we may say, from the ground and giving the results I have reached from my personal observation.

Friction of Wheel Upon Ground

A series of laboratory tests on full-sized wheels, which checked very well with work on the machine in the open, resulted in the following conclusions:

(1) The maximum drawbar pull is a definite function of the weight per inch of width. Weights used varied from 10 to 200 lb. per in.; the ratio of maximum possible drawbar pull to total weight on the wheel was constant for that range. This was found to be true whether the ground were wet or dry.

(2) On sandy ground the drawbar pull available with a smooth metal wheel is about 30 per cent of the weight on the wheel.

(3) On damp, sandy ground the maximum drawbar pull is greater, being about 43 per cent of the weight, and under some conditions even slightly higher.

(4) Cleats increase the maximum drawbar pull only insofar as the soil resists shearing; that is, the cleat carries a section of the top soil and slides it against the soil below the edge of the cleat. Experiments indicated that this was practically independent of the depth of the cleat, depending solely on the shearing strength of the soil at the depth of the cleat edge. In some cases the shallower cleat pulled more than the deeper cleat because the roots in the sod were not cut off and advantage was taken of their shearing strength.

In some cases subsoil may be more tenacious than the top, but this is unusual. It might be supposed that when the top is sheared from the subsoil, the soil at the *back of the wheel* would support it if the section were deep, to a greater extent than if it were shallow. This does not appear to be the case because of the lifting action of the back of the wheel which tends to eliminate the support. This is particularly noticeable on hills when the wheel is stressed to the limit. It was found that a cleat inclined forward at an angle would improve this condition somewhat. In going uphill the cleat enters the soil almost horizontally, acting like a step and tending to lift the weight off the wheel. On leaving, the cleat stands almost vertical and causes less friction and loss of power. An inclination of about 30 deg. was found to be the most satisfactory on a 6-ft. wheel. This arrangement tends to self-clean to a great extent. Setting the cleats at an angle of 30 deg. to the axis of the wheel also helps this cleaning effect by a slipping action. The shearing strength of the soils I tested appeared to vary from 5 lb. per in. of width in dry molding sand to 75 lb. in loam or sod. No doubt, tough sod or gumbo may prove even stronger than this.

From these facts it becomes evident that weight is the only means of obtaining a tractive effort of 40 per cent of the weight of the machine under bad conditions in dry ground or sand, as cleats will be of little use. Wheels 72 in. wide would give an added pull due to the use of cleats of only 360 lb. for loose ground. Weight is therefore practically the sole reliance for traction in sand or very dry loose ground. In sod or damp ground 72-in. wheels would ordinarily give 4000 to 5000 lb. pull from the cleats alone, and the light machine with only sufficient weight to hold the cleats down would show good results.

Power Required to Propel the Tractor

Certain observations in operating a tractor led me to believe that the rolling losses were greater than I had anticipated, and I began a study to determine the laws governing the subject.

I took a watering cart having 46-in. wheels with 8-in. flat tires, and removed the tank, leaving only the stripped chassis. This was towed by a motor truck over several kinds of ground, and the drawbar pull was measured by spring scales. After this I replaced the body and filled it successively with various amounts of water to increase the load per inch of width on the wheels. Four types of ground surface were tested: (a) hard macadam road; (b) dry short clover; (c) sod long grass (gravelly loam); (d) dry plowed land, such as would ordinarily be harrowed, plowed 24 hr. previously.

The drawbar pull was found to increase rapidly with the weight per inch of width on a regular curve on all but the hard macadam road, where the increase was only slight. This was to be expected, as the increase in pull is due to the work done in crushing the ground surface, or, in effect, continually pulling the wheel uphill. The limits selected for the tests were those under which a

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tractor can be built. The curve, Fig. 1, shows the ratio of the drawbar pull required for propulsion to weight plotted against wheel width in inches. The speed used was about 2 miles per hr. I could notice no marked difference in the pull from 1 to 3 miles. On some previous experiments I had observed that the rolling friction of wheels of various sizes varies inversely as their diameter and I believe that this fact can be used with the chart as a basis for design. It would certainly be valuable to have these observations extended further to furnish a more certain basis for calculation. I have yet to see any precise data on this subject.

The value of these facts is obvious. If a tractor has too great weight per inch of width of wheel and gets on soft ground, as it must do in harrowing, etc., the power consumed in rolling friction becomes a large percentage of the total developed and the drawbar pull is proportionately decreased. There is every advantage in increasing the surface and reducing the weight. Tractors with a weight of over 200 lb. per in. of width of wheels that are even 6 ft. in diameter become very inefficient on soft ground. It is here that the "caterpillar" shows to best advantage, and if it were not for its complication and high maintenance cost, it might prove somewhat better than the wheel for many purposes. When properly proportioned, however, the wheeled machine can be made to give a very high efficiency, as I will show later on, and since wheels can be made to last the life of the machine without expense or renewal, their design is worthy of very careful study.

Weight on Wheels

Considering a four-wheel machine, with the rear wheels as drivers, it is evident that no matter what form of mechanism is used to apply the power to the wheels, the maximum drawbar pull multiplied by its lever arm ($P \times a$) cannot exceed the weight on the front wheels multiplied by a lever arm equal to the distance from the center of the front to the center of the rear wheel ($W \times b$), Fig. 2. The maximum pull which can be applied will be only that required to lift the front weight off the ground. If the machine is on a hill, this weight will be reduced as the cosine of the grade angle. In other words, when we apply power to the rear wheels we are in effect transferring part of the weight of the machine from the front to the rear wheels.

From the preceding curves it is easily seen that the increase in the weight on the rear wheels would increase the weight per inch, and so increase the rolling friction. In the first machine I designed this was very marked, as I increased the weight on the rear wheels 3500 lb. by applying full engine power, and increased the rolling friction from 20 to 25 per cent on plowed ground. The machine, therefore, did not show the drawbar pull it should from power developed by the engine. It took me several months to find the cause of the losses, and I was not absolutely certain until I had actually brake-tested the machine on the rear wheels under full power and found that the mechanical losses from the engine to the ground were only 16 per cent and did not increase materially with the lower gear ratios or increased power. Besides these serious frictional losses, very few machines in this country are proportioned in such a way that the full engine power can be applied on the low gear on any reasonable hill and not raise the front of the tractor off the ground. This interferes with the steering, and in many cases the tractor will turn over if the operator does not shut off the power. This is dangerous, and the full power of the tractor is not available. The utmost care should be exercised in having the weight on the front exactly right for the work it is desired to perform, and any failure here is inexcusable.

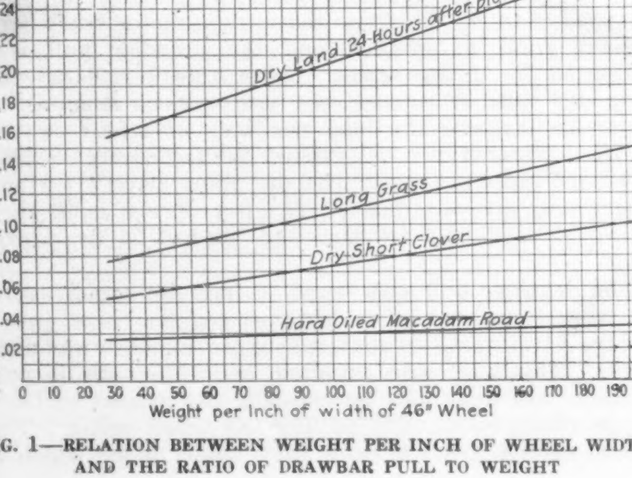

FIG. 1—RELATION BETWEEN WEIGHT PER INCH OF WHEEL WIDTH AND THE RATIO OF DRAWBAR PULL TO WEIGHT

If we consider a machine driving from the front wheels, the reverse is true, and the application of power decreases the weight on the drivers and reduces the possible drawbar pull which can be delivered with a given weight of tractor. This type is therefore inefficient and will eventually be abandoned. The rear drive type can be proportioned so that nearly all the weight of the front can be carried by the rear wheels when the full engine power is applied, leaving only enough weight for proper steering. This form will then develop the greatest possible pull for a given total weight.

Over-All Efficiency

By using anti-friction bearings, good lubrication and cut gears of good design, it is possible to reduce the frictional losses from the engine to the rim of the rear wheels to less than 16 per cent of the total power developed. I found on brake test of one machine that the losses with a gear reduction of 172:1 on third speed were not materially greater than the losses at 52:1 on the direct drive, where they proved to be 12.5 per cent. This is understood when we consider that we are only interposing two additional pairs of gears, the gears being well oiled and of ample size. As these gears are running in oil when the tractor is operating on the direct drive, they have under these conditions the friction of the oil in the box, which is very considerable, gearbox-oil friction with heavy oil being about 3 hp., and about ½ hp. with light oil. To the mechanical losses must be added the rolling friction on the ground. As seen above, this may vary from 8 to 20 per cent with properly proportioned wheels.

In testing the engine on the brake I used a vacuum gage on the inlet pipe and noted the suction at various loads at the governed speed. When the engine was put on the tractor again I used this gage, which gave me the actual horsepower of the engine within a very small error, as the engine in both cases was operating at the same governed speed. A speedometer was employed that gave the feet of advance per minute of the tractor on each gear. A spring dynamometer was applied to the drawbar, giving the reading in pounds. It was a simple matter to multiply the pounds pull by the feet of advance per minute, get the delivered horsepower and com-

pare it with the actual engine power. It was found that on good sod with short grass the efficiency of the machine was about 75 per cent, i.e., the drawbar horsepower divided by the engine horsepower was 0.75. On plowed ground, harrowing or cross plowing, it was about 65 per cent, and never fell below that figure. When we consider the low efficiency of touring cars and trucks this seems a very satisfactory result on soft ground, and one which the engineer would scarcely expect to exceed.

Fuel Efficiency and Gear Changes

The internal-combustion engine of conventional type has one serious drawback; if it is overloaded it stops at once, unless it is operating at a speed such that it is beyond its maximum torque point, and slowing down allows the torque to increase. In this case the normal load of the engine can be slightly increased at the expense of the speed. These being operating conditions, it is evident that on a tractor the engine must be regularly operated below full load, or a specially hard pull on the plows or a slight hill will cause it to stop or necessitate a change of gears. It is, therefore, the usual practice to gear tractors in such a way that the engine is operated only at a small percentage of full load, say 40 or 50 per cent. This provides the safety factor for overloads and for defective oiling and cooling, as the overloads may prove to be of short duration. The curves, Fig. 3, taken as an average of tests made during several years in my

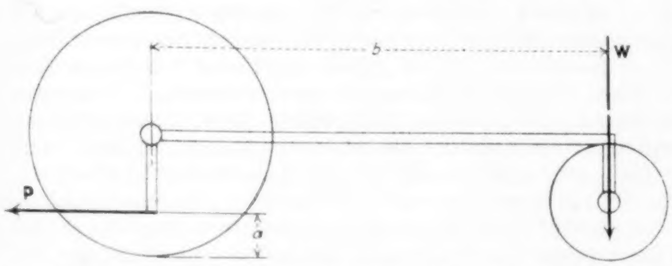

FIG. 2—DIAGRAM OF THE RELATION BETWEEN THE MAXIMUM DRAWBAR PULL DEVELOPED BY A TRACTOR AND THE WEIGHT ON THE WHEELS

own laboratory of a number of good engines, show the best fuel economy at various percentages of full load, all tests being made at the constant speed of 1000 r.p.m. The lower curve shows the best fuel economy I have yet obtained when operating conditions were at their best. The curve marked 1 may be taken to represent ordinary practice in commercial tractors. As we cannot operate continuously at full load and best economy, it is interesting to see just how far below this point we can go before the increase in fuel becomes very marked. It will be noted that we should not go below 60 per cent on an ordinary engine, while on the best ones under good conditions a fuel economy of 0.66 lb. per hp.-hr. can be obtained at 50 per cent load. Tractor engines should not be used below this point over long periods. Another point to be noted is the absolute necessity of having the engine oiled and cooled so that it can run all the time at 50 per cent to full load, if we are to secure good fuel economy. A tractor, however, should work at certain specified speeds for best results on the tools used, and the pull of these tools will vary as much as 100 per cent in the same field, not to mention hills and variations of rolling resistance.

How, then, are we to get good fuel economy with two changes of gears? In my original machine I had three speeds; the high speed was reversed for road use and

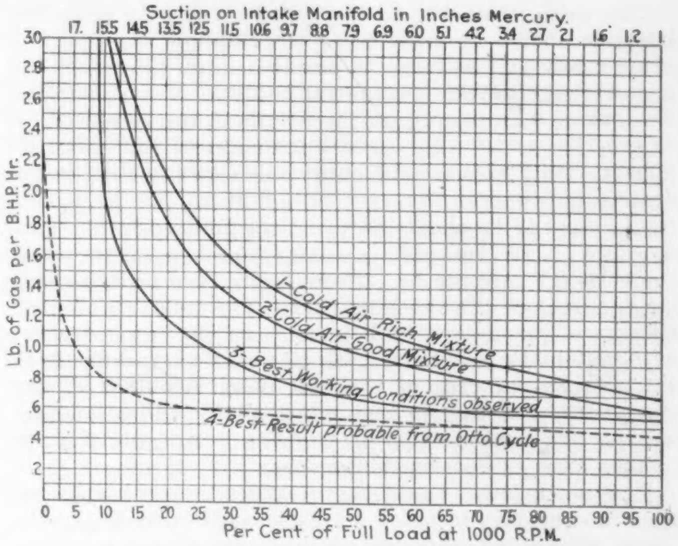

FIG. 3—FUEL ECONOMY AT VARIOUS PERCENTAGES OF LOAD AVERAGED FROM A NUMBER OF TESTS

could not be used in the field, leaving only two speeds for actual work. I have been obliged to adopt four speeds so as to have three available speeds for field work. These are respectively 1, 2 and 3 miles per hr. at 1000 revolutions of the engine. The latter is governed to 1200 r.p.m. In this way it can be used under the proper load conditions all the time for best economy.

I use a suction gage on the inlet pipe as an indicator of the engine load, and have this marked so that the driver knows that it should not read over 8 in. of mercury suction, which is 50 per cent load. This point was selected as the minimum load under which the engine will operate economically. If the suction rises above this point, the driver should shift to a higher gear and load up the engine further. This is just the opposite of touring car practice, where we shift gears because the engine is overloaded and needs relief. On the tractor, the heavier the load, the better the fuel economy will be, and the better the financial result, provided the engine is able to stand this load. We must build engines capable of holding their full power, and a cooling system that will keep them within reasonable ranges of temperature under all conditions, if we are to secure economical operation.

Size of Machine

The size of the machine is a subject of supreme importance, and I am in the unfortunate position of being obliged to differ with the general opinion that the small tractor is the future economical unit.

The expense of operating may be divided into the three main headings of operator's wages, fuel and oil, and maintenance charges.

The driver on the small machine is paid as much as on the large. The maintenance charges for actual repairs on the larger machine, if properly constructed, are not materially greater than on the small machine. The only difference here will be in the interest and depreciation items. The cost of fuel and oil will prove to be the main difference in expense. This, however, is much lower per acre on the large unit than on the small; therefore, the operating expense per acre on the large machine will be less than on the small machine, primarily because the driver's wages will prove to be at least one-third of the whole cost.

The evident answer to the question of how large the machine should be made is as large as one man can operate easily and as large as can be run on the roads and pass through gateways and over bridges. I consider that a width of 9 ft. overall on the wheels is the outside limit in practice, and a total operating weight of 14,000 to 16,000 lb. With this weight eight plows can readily be operated, plowing at the rate of 2.8 acres per hr. This would seem to be about the practical limit for a one-man machine. The two-plow machine, on the same basis, would do 0.7 acre per hr. These figures are for the regular progress of the machine over the ground, making no allowance for delays and turns. In factory practice, with automatic machinery, it is found very difficult to get over 80 per cent of the best possible output of the machinery, and this is true when all the oiling and repairing are done outside of working hours and the operators have skilled superintendence.

How much lower ought we to expect the percentage of time in operation to be on a tractor subject to delays for oiling, filling and adjustments, not only of the machine, but of the plows and tools, stopping for corners and turns, shifting gears, etc.? An output of 60 per cent of the theoretical would be very good indeed, and I fear that it often proves to be very much lower. If we take 50 per cent of 7 acres for a day's work we get $3\frac{1}{2}$ acres on an average, and this is more than is usually obtained over any long period with the two-plow machine.

At the present time there are no eight-bottom tractor plows on the market which can be worked satisfactorily from the tractor by one man. I have made a set for my machine which works fairly well, but it will be some years before these devices can be perfected. Four-bottom plows are now to be had which with a few changes can be made to work fairly well from the tractor. It is probable that at the present time, taking into consideration all the facts of economy, first cost and ease of operation, the four-plow machine is the best suited to build. This is especially true when we consider that it is possible to design a four-plow machine with substantially no side draft and operated with the tractor wheels of ample size running on the unplowed ground.

I have just had the opportunity of getting figures from three of the best standard makes of machines, working in Orange County, New York, on difficult land. They were operated for a whole season by a township committee. The machines have averaged about an acre a day each during the season. The same money invested in horses would have done more and a larger variety of work. The cost per acre has been much greater than with horses, and the three machines all need rebuilding at considerable expense. With a capacity of 20 to 30 acres a day, we have a good chance to get 10 or 15 acres regularly. In my view, it is not the first cost of the whole outfit that will be the deciding factor, but the investment per acre worked per year. If the large and more expensive machine will do the work more cheaply, and at the same time show a lower investment per acre, it will be the one to survive in the long run. Small machines are advertised to pull mowing machines, reapers, cultivators, etc. It seems rather a step backward to be pulling these machines by a means costing at the lowest estimate from \$5 to \$8 a day to operate, when we can do it with horses at not over \$4 to \$4.50; or to be running a thresher and silo cutter with a machine costing between \$900 to \$1,500, when we can do the same work with a stationary engine costing between \$200 and \$300; while the tractor could be plowing and doing expensive work on the farm. My view is that the small tractor is rarely needed at all for the small farm. On any farm up to 100 acres the work can be done with horses more cheaply and better, especially if the farmer uses the right crop rotation and carries enough live stock. The small tractor would only be a source of expense, as he would certainly have to keep horses as well. The tractor should have at least 200 acres to cover a year, and better still, 500 or 1000. On this large output a skilled man can be employed and the machine properly worked. The tractor and farm tools that the farmer will have available for some years to come are certain to be the kind of machinery that should not be placed in the hands of the untrained.

Conclusion

In closing, I would like to add a word as to what can be expected in the future as to the maintenance and durability of good tractors.

A motor-truck designing experience of twelve years has taught me that a well-made, properly designed gear-box and clutch will last from five to ten years with no repairs, and I see no reason why we should not expect even better results on tractors if we proportion the loads properly, especially as the tractor will not operate more than 120 days a year. The wheels on my machine show no wear after two years' work, and I am sure they will last the life of the machine, with the possible renewal of cleats every four or five years. The frame, riveting and steering-gear can be made absolutely durable. The two places in which we will have excessive wear are on the final drive to the rear wheels and on the engine. The main drive to the wheels is not durable on any machines I have seen. On my own it has lasted two years and requires renewal. It will be replaced with a drive of better proportion and materials, and better protected from dirt; I hope to get four years out of it at least. This can, no doubt, be further improved. On the engine the main source of wear is the dust in the cylinders. Even with a fine muslin dust filter 8 sq. ft. in area the pistons and rings were badly worn in 100 days' running. A perfect air cleaner is an absolute necessity for tractors. Those I have seen advertised thus far are all of types which will not really remove the finest particles of dust, which do the actual damage to the engine, and they cannot eliminate the trouble. If we succeed in protecting the engine from dirt, there is no reason why we cannot get ten years' service out of it, with a regular overhaul every year or two. I know of motor-truck engines in operation today which have been in constant use for over ten years, and I firmly believe we shall make a tractor in the near future with a very low maintenance cost and a very low operating cost per acre. I believe we may hope to plow for about 60 cents per acre, on good land, this including all charges, as against \$3 to \$4.50 with horses. If this is accomplished, the tractor engineer will have done his bit toward solving the food problem.

The Metric System

FREDERICK A. HALSEY* presented at the last meeting of the American Society of Mechanical Engineers the results of an investigation to ascertain, from replies to about 500 questionnaires sent by American firms to firms and individuals in Latin-American countries, information concerning the systems of weights and measures used in retail buying and selling, in the clothing trade, land measures, industries, farm products, rail and ship transportation. He stated that nowhere is the metric system used exclusively, and that in countries that have adopted it, the English system also is used, together with the original Spanish units which differ little from the corresponding Anglo-Saxon ones.

The aim of the author was to disprove: (1) That it is an easy and simple matter for a country to change its system of weights and measures; (2) That the adoption of the metric system does away with confusion; (3) That this system is in universal use except in the United States, the British Empire and Russia; (4) That we must adopt the metric system if we desire to do business with Spanish America; (5) That the adoption of this system saves time in schools; (6) That it saves time in calculations; (7) That the persistence of old units in metric units is a persistence of names but not of things; (8) That we will use the exact metric equivalents for English sizes.

The paper evoked lively discussion. James Hartness said that while he wished to be looked upon as a pro-inch man, and felt that the industrial worker preferred to adhere to the inch system, he had a high regard for the scientist who is naturally in favor of the metric system. Realizing that ultimately one system of weights and measures must prevail in international dealings, and in view of the present preponderence of Anglo-Saxon influence he thought that one way out of the difficulty is to make the meter equal to 40 in., as had been proposed in the past.

Howard Richards, Jr., secretary of the American Metric Association, New York, emphasized the great simplicity of the metric system of measures. He challenged any one present to give him off-hand the length of the side of an acre square, and elicited three replies, all differing and none exact.

Mr. Richards quoted Edwin M. Herr, president of the Westinghouse Electric & Mfg. Co., East Pittsburgh, as saying that the Westinghouse company could make the change to the metric system by cooperating with the other engineers.

Henry D. Sharpe considered that the work of Mr. Halsey disproved the claim that there is urgent need of adopting the metric system to hold foreign trade.

A 40-In. Meter Proposed

Ralph E. Flanders offered the following motion, which was seconded by Mr. Hartness:

> *Resolved:* That it is the sense of this meeting that the possibilities of the changing of the meter to forty (40) in. be brought to the attention of the Council of this Society, and that the Council be asked to consider such action as may lead to the adoption of this change by the metric countries.

Luther D. Burlingame found himself at variance with Dr. S. W. Stratton, director of the U. S. Bureau of Standards, who had contended that the metric system is the most popular one in Latin-American countries. That the English system is acceptable to twenty Latin-American countries, was according to Mr. Burlingame, proved by the fact that the total trade between the United States and those countries during the fiscal year 1917-1918 exceeded that of the year 1913-1914 by $1,000,000,000.

Adolph L. De Leeuw thought a 40-in. meter would add greatly to the confusion. He felt that the question of change of system had not been considered with an open mind. He said we should ask ourselves three questions: (1) If we had to choose a new system, would we select the metric or the English? (2) How much would it cost to change to the metric system? (3) Would the benefits be commensurate with the cost of changing?

Samuel S. Dale, editor of *Textiles*, Boston, gave a lengthy discussion in favor of making the English system the universal standard. He showed that the present Spanish tables give the same multiples as the English, while the units themselves vary only 1.4 per cent from our units. He stated that in Russia the linear measures are based on the English inch and foot. The Chinese foot is equal to 12.6 English in., but is divided decimally. The Japanese foot is 11.93 English in. Even before the war, 70 per cent of the world's industrial energy was based on English weights and measures, and the situation now at the end of the war precludes the adoption of any but the English system as a world standard. He called attention to the fact that British committees reporting to Parliament regarding British policy after the war, did not recommend the compulsory adoption of the metric system, but held that a universal system would have to be acceptable to the British Empire and to the United States, and that in the meantime a simplification of the British system should be effected.

Henry M. Hobart said he believed that a 40-in. meter would be equally unpopular with all parties concerned; also that few realize to what a great extent the metric system is already established in the world. He was convinced that while working with the metric system in Germany his efficiency was higher than when he used the English system in Great Britain and the United States. He said it is not true that the British system is used to any considerable extent in Germany, but that the metric system is in use almost universally, as an inspection of textbooks will show. He thought this was the reason Germany could produce more efficiently and cheaply than the English-speaking countries.

Charles W. Johnson prefaced his remarks by stating that his viewpoint was that of a machine-shop man. He reasoned that if it is true that it takes a year longer to educate a child in the English system than in the metric, that is an advantage because it makes him think, something he is supposed to learn how to do. Mr. Johnson wished to go on record as being decidedly opposed to the exclusive use of the metric system, for the sake of the operating men in industry. He said he had as yet discovered no advocate of a complete adoption of the metric system among industries manufacturing large varieties of complicated products where repair parts must be supplied. He thought the cost of retracing 500,000 drawings and converting inches into

*Commissioner, American Institute of Weights and Measures.

millimeters would be prohibitive, and bring no corresponding advantages. The method preferred now is to use even figures or simple fractions, because they are easier to work to than extended decimals. To employ metric units on new work only, he thought to be impracticable because of the mixed system of machine tools which would be necessary; besides, many standardized pieces are incorporated in new designs and the result would be great confusion. He referred to an article in the September 1916 number of *Machinery*, entitled "The Metric Agitation," as being a clear and honest exposition of conditions as affecting the machine shop.

Mr. Johnson added that if Latin-American business men prefer that shipping papers give weights and dimensions of packages in metric units, there is no objection to that, and that metric outline and foundation plans can be readily supplied, but he could not see why the parts themselves should be fabricated according to the metric system. He said it would be most inopportune to make such a change now, because in the years following the war, with taxes high and labor conditions disturbed, it would be necessary to practice economy and efficiency more than ever.

Charles E. Skinner, representing the same company as Mr. Johnson, was in favor of having but one international system. It did not matter which, but as to adopting the metric system, he was opposed for the same reasons Mr. Johnson had given, because he said the vice-president of the Westinghouse Electric & Mfg. Co. had estimated the initial cost of retracing its 1,000,000 drawings to be at least $1,000,000, with no compensating advantage. Furthermore, the English-speaking peoples now occupy the dominating place in the world, and they can retain it better by adhering to their present system.

D. E. Lauderburn said he had been practicing forest engineering since 1907, and had cruised timber in Cuba and Santo Domingo. He found that the fundamental standards in these countries were metric, that the governments were using metric weights and measures, and that the surveyors used the metric system with the additional unit caballeria, which represented 42 metric hectares, the same as in Mexico. He exhibited a booklet published in 1917, in Spanish and English entitled "A Brief Statistical and Geographic Review, Including Revised Map of the Dominican Republic," on which map the only scale used was: 1 mm.-1 km., and thought that especially for drawing and reading maps metric measures are a great convenience. He also exhibited a 28 by 63½-in. chart, published by the Cuban Department of Agriculture, Commerce and Labor at Havana, Dec. 15, 1911, and signed by four government officials, giving all required technical data regarding 367 different kinds of wood. On this sheet the height and diameter of trees were given in meters, and the structural strengths expressed in kilograms per square centimeter.

After Mr. Sharpe had made further opposition to the resolution recommending consideration by the A. S. M. E. Council of the 40-in. meter and Mr. Flanders once more had urged its adoption, Mr. Sharpe's motion to lay this resolution on the table prevailed by a rising vote.

Mr. Halsey in closing the discussion, stated that the proposal for a 40-in. meter had been made frequently, the first time by Joseph Woodworth some fifty years ago, when little difficulty would have been involved to modify it. He said that his knowledge of the New York schools had convinced him that the study of weights and measure, instead of requiring a year, never required more than three weeks. "The judgment of the world is that the metric is not the better system, because the great majority refuse to use it."

He referred to France, where the metric system was first promulgated by law in 1792. In 1812, under Napoleon's reign, the law was repealed and people were allowed to return to their old system. He said this showed what the French people thought about this system; and that in South America they did not use it because they did not like it.

A Uniform Decimal System Urged

John H. Wigmore, Office of the Provost Marshal General, Washington, wrote that his attitude is based on the conviction that a system of weights and measures uniform throughout the commercial world, and decimal in its nature, is the obvious requirement; that he considers the Anglo-Saxon system unscientific and impractical, and that the undoubted inconveniences of transition would be more than made up by the great convenience and economy of future use; also that his opinion was strengthened by information he had received showing that in the engineering schools of the United States, the metric system is used in practical work.

Representative James L. Slayden, Chairman of Committee on the Library, House of Representatives, Washington, wrote that he feared it would take considerable effort to secure the introduction of a really scientific and, when known, convenient system of weights and measures. He said that stupidity, inertia and momentary selfishness all operate to defeat the general adoption of the metric system in the United States. During his long and intimate acquaintance with affairs in Mexico, he had been greatly surprised to see how easily even the natives acquired a knowledge of the metric system, which was irresistibly making its way.

John B. Moore wrote that so long as Great Britain and the United States continue to do business in Latin-American countries in the English units, these units are bound to be retained by those countries, unless their use be prohibited or penalized. He said that a uniform decimal system appeals very strongly to one's sense of convenience. On the other hand, he fully appreciated the fact that certain vested interests would be temporarily affected by the change, as well as the fact that changes are not generally desired by those who are not actually suffering from existing conditions.

H. J. Bingham Powell wrote that during his several years of experience as a civil and mechanical engineer in Peru and Bolivia, he found that in his line of work the metric system prevailed, but that manufactured articles imported from England and the United States, such as pipes, valves, cement, etc., were designated by their original units, inches and barrels, and timber in board feet. However, in workshops inches were in use rather than millimeters. In hydraulic calculations liters or gallons were used indiscriminately, the governments not attempting to make the metric system compulsory, since it would only inconvenience the people, who were satisfied with the mixed system of measurements.

William Jay Schieffelin, New York, whose drug business was founded in 1794, wrote that Mr. Halsey's report shows the chief obstacles to the wider use of the metric system in South America to be their business

dealings with the English-speaking people. It certainly is not surprising to note the slow adoption of the metric system by the illiterate and conservative natives, but that is no reason for the people of the United States to be equally slow and shut their eyes to the lessons of the past year, which demonstrated how quickly the whole people of this country can be informed as to an issue and within a few weeks voluntarily cease using certain commodities, or change their habits of life, if it is for the country's good. He thought it ought not to take long to show a person having business dealings that a simple decimal system will save time and mistakes in merchandising. In a wholesale drug house 30 per cent of the time of clerks could be saved if the weights and measures were metric. The fact that our blindness is not only hurting us but our southern neighbors ought to make us doubly anxious to put the reform into force.

Dr. A. E. Kennelly wrote that those who had visited Continental Europe and the Latin countries knew from experience that the metric system is in universal use there and that if it were not satisfactory efforts would be made for the adoption of something better. Further, that the world is too small to permit of an indefinite continuance of a plurality of official systems. Sooner or later one system must prevail. When a simple and internationally recognized system operates alongside a complex and local system, it is only a question of time when the complex system must disappear. Practically all scientific and electrical work is conducted and recorded in the metric system in every country. The American Institute of Weights and Measures could perform a great service to America and the world by studying without bias means for bringing about international uniformity. That this could be done without abandoning existing plants, machines or apparatus was evidenced by the experience of other countries, the adjustment would only involve considerable readjustment of ideas, records and drawings. How to reduce this expense to a minimum, and how those burdened with the cost of changing over shall be protected and insured, were questions worthy of attention.

THE BATTLE OF THE MARNE

HAVING arrived at Chantilly in the early part of September, after the withdrawal of the French-English Army, von Klück hesitated to pursue his march direct upon Paris. In fact, it would have been imprudent to strike the entrenched camp that General Galliéni had organized around the city and defended by a body of excellent troops under the command of General Maunoury. The attempt would have been perilous to a supreme degree while the rear of the French Army was within a short distance, perfectly intact, and with an unbroken line. A counter thrust from the French Army would mean complete disaster to the Germans.

They therefore resolved to destroy if possible, and failing this, to disrupt the armies of General Joffre and scatter them in all directions, leaving on the south an open road to Paris. The movements of the Germans inclined toward the east, withdrawing from Paris. The change of direction began on Sept. 3. On the next day the Germans occupied the Marne, Lagny, Meaux, La Ferté-sous-Jouarre, Chateau Thierry and Epernay. On the evening of the 5th they had advanced still farther. Their advance guards extended beyond Coulommiers in the direction of Provins, but in making a flank movement toward the camp entrenched around Paris a gap was formed in the attacking column. Here Galliéni immediately launched the Army of Paris under Maunoury.

It was the hour the commanding general had waited for, had foreseen and had anticipated. At once he gave the command to halt, about face and make preparations for the battle, a series of simultaneous battles engaging 2,000,000 men on a front of less than 300 km. (about 186.5 miles). The attack begun by Maunoury on the 5th continued for several days. He had the entire army of von Klück at a disadvantage, who, seeing the danger that threatened him, recrossed the Marne and turned back toward the north. Maunoury's men held their ground with heroic determination in the valley of the Ourcq during those three terrible days, Sept. 6, 7 and 8. They were joined by the English under Marshal French, who had retaken Coulommiers and had carried by assault the Pass of Petit-Morin. In passing from the west to the east we have, first Frenchet d'Esperey, who on the 8th was at Montmirail and Vauchamps; then Foch, who had before him the Prussian Guard; Langle de Cary, who was on the Ornaine, on the heights of Vitry-le-Francois, and Sarrail at Verdun.

The struggle took place almost on this very spot, but now it is no longer the Germans who execute the outflanking maneuver. In their turn they risk being outflanked, and at the same time are being turned back on our left wing by Maunoury and on our right by Sarrail. On Sept. 9 they made a desperate effort to break our line in the center, but instead they were broken there themselves. On the morning of the 10th they were forced to retreat. They were completely routed during Sept. 10, 11 and 12. Foch by a daring blow overthrew the Guard in the Marshes of Saint-Gond.

Ten French victories interlocked each other. The line of the Marne was regained and other territory added. Compiégne, Soissons, Rheims and Chalons were evacuated in the confusion of flight. The French pursuit continued beyond the Aisne, where the Germans entrenched themselves in almost impregnable positions preparatory to an advance. However, they might even then have been chased to a finish if the enthusiasm of our valiant troops could have been maintained, but they were totally exhausted from a whole week's fighting without having had any rest. You must remember that Sept. 12 was marked by the definite check of the assault which since Aug. 22 had been carried forward steadily by the Germans under the very eyes of the Kaiser himself, against the Grand-Couronné de Nancy, which, by the way, was defended in a masterly manner by Castelman.

The victory of the Marne is one of the most memorable events in the history of the world. It saved France and the entire world from Prussian domination.—Translated from Hachette's "Historic Résumé de la Guerre," 1917, by Laura de Russy Berry.

Air Flow Through Poppet Valves*

THIS discussion deals particularly with the merits of inlet valves in pairs, as compared with the single inlet. The experimental data presented afford a direct comparison of valves singly and in pairs of different sizes, tested in a cylinder designed in accordance with current practice in aviation engines. Unfortunately, necessity limited the investigation to measurements taken under conditions of continuous flow.

The investigation was undertaken after a wholly unprofitable search for accurate information upon the comparative flow characteristics of single and double inlet valves, based upon actual measurement rather than upon some hypothesis, itself largely a matter of opinion.

By way of preliminary analysis, the application of the law of geometrical similarity presents a strong case for valves in pairs. For example, at a given pressure drop and the same lift, one valve would require a diameter of 4 in. to provide an area of opening equal to that of a pair of valves each of 2-in. diameter. The superficial area of the one 4-in. valve is twice the combined area of the two 2-in. valves, and if opened against a pressure in the cylinder, this is a measure of the comparative forces involved. The 4-in. valve would weigh four times the combined weight of the 2-in. pair, and the necessary spring tension would differ in that proportion, for the same lift and the same engine speed. It may be noted here that, while the above is correct upon the assumption of geometric similarity, the effective valve areas differ from the actual, as the coefficient of efflux varies at different lifts; also, that the weight of a well-designed valve increases somewhat less than the third power of the diameter would indicate.

L. H. Pomeroy, in a discussion which he states is wholly analytical, reaches a conclusion decidedly at variance with the ones above. Briefly stated, he assumes that two valves of 2.83-in. diameter should be substituted for one of 4-in. diameter (equal cross-sectional port area which requires that the smaller diameter be 0.707 of the larger diameter) and that the valves in each case are lifted 31.65 per cent of their respective diameters. He then computes the hydraulic mean radii for the two cases, applies the laws of friction, and reaches the conclusion that the two valves would have a frictional resistance 39 per cent greater than the single valve.

The contrast is sharp. The tentative conclusion geometrically derived is that two valves of *one-half* the cross-sectional port area and equal opening area, as compared to the single valve would afford the same flow. Mr. Pomeroy's tentative conclusion is that two valves having the *same* cross-sectional port area as the single valve, and the same opening area with a lift 0.707 that of the single valve, would have a frictional resistance 39 per cent greater, and therefore less capacity. This discrepancy seemed to afford ample ground for experimentally determining the relative flow in similar combinations of valves.

Apparatus

The apparatus used consisted principally of a centrifugal blower, a model cylinder, and U-tubes for measurements of pressure.

The blower was one of special design with a balanced rotor of 11.25-in. diameter, composed of ten forward curved blades. An electric motor furnished the power, rheostat control permitting speeds from 3000 to 6500 r.p.m., corresponding approximately to pressures of 9 to 32 in. of water. The number of impulses varied from 30,000 to 65,000 per min., affording practically continuous flow. The blower was connected to the cylinder with rubber hose, care being taken that the alignment of the hose remained perpendicular to the face of the cylinder at the point of entrance throughout the tests.

The cylinder is shown in longitudinal cross-section in Fig. 1. The cylinder head was carved out of white pine and carefully finished in accordance with dimensioned drawings. At the entrance end, the passages leading to the valves were cylindrical in form with their axes perpendicular to the cylinder axis and of 2.5-in. diameter; the passages then curved as shown to the ports. The approach to the large valve, which had a diameter of 2.5 in., was circular in cross-section at all points. The approach to the pair of valves on the opposite side of the cylinder became narrower in the plane of the cross-section shown, and widened laterally to divide smoothly, about 1.5 in. from the ports, into two passages of 1.75-in. diameter. The angle between the valve axis and the cylinder axis was 15 deg. No valve guides or bushings extended into the passages.

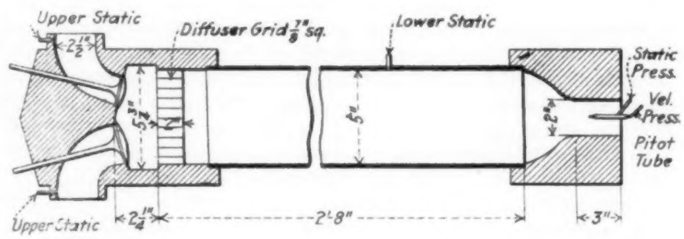

FIG. 1—CROSS-SECTION THROUGH CYLINDER MODEL

The diameter of the counterbore was 5.75 in. and of the cylinder proper, 5 in. The valves were seated with a bevel of 30 deg. in the two planes forming the cylinder head. The diffuser shown was constructed of thin brass soldered together and inserted so as to divide the whole area of the cylinder at that point into rectangular passages about 0.875 in. square and 2 in. long.

The jet at the opposite end of the cylinder was likewise carved out of white pine as shown, and was connected to the cylinder head by a length of 5-in. wrought-iron pipe, smoothly galvanized inside, used to obtain sufficient length for rectification of the air current. Gaskets and shellac were used at the joints and the assembly drawn together with four long bolts extending from end to end, outside the cylinder.

In addition to the single valve with a diameter of 2.5 in. and the pair of valves with diameters of 1.75 in. already mentioned, another pair with diameters of 1.25 in. was tested. False seats were used with this smaller pair, consisting of turned hardwood rings, carefully fitted to the 1.75-in. seats and beveled to receive the smaller valves. These false seats obviously left a circular shelf or projection 0.25 in. wide immediately above the ports. As a matter of interest, two readings were taken with these shelves projecting above the port, but before running off the main test on these 1.25-in. valves, the

*From report of National Advisory Committee for Aeronautics. (Clarke Thomson Research)

lines of the passages were smoothed off by filling in above these projections with putty.

The valves were all designed on similar lines with the exception that the smallest pair had stems of 5/16-in. diameter, to fit the guides used for the larger pair, this dimension being 40 per cent larger than true proportion dictated, equivalent to a reduction of 0.022 sq. in., or 1.8 per cent of the port area of the smaller pair.

The Pitot tube shown in the jet in Fig. 1 was clamped in position at the axis of the jet throughout the tests, velocity readings being taken as described later. The dimensions were 3/16-in. outside diameter and of about 2.5 in. length. The impact end was gradually rounded, and the static holes were four in number, about 0.02-in. diameter, smoothly perforating the outer walls.

A static tube of 0.125-in. diameter penetrated the central portion of the cylinder, reading static pressure of the air column after passing the valves and the diffuser. This is for convenience termed the "lower static."

Static tubes of the same diameter also tapped the flow where the air column entered the passage leading to the valves. These are for convenience termed "upper static," only one being used at a time, as indicated by its position with respect to the valves. All statics were slightly rounded on the inner periphery, and the end kept flush with the inner surface of the cylinder or passage, and so located as to be perpendicular to the direction of air flow.

The upper and lower statics were connected to the two legs of a U-tube to read the pressure drop through the valve directly and also connected to other U-tubes to read the upper static and lower static head separately.

All U-tubes had an inside diameter of about 0.25 in. and were vertical with the exception of one, which was inclined at a slope of 10 to 1 to read with greater accuracy velocity pressures of 3 in. or less.

A centigrade thermometer was clamped with its bare bulb in the air jet at a point about 1.5 in. outside the apparatus. A similar thermometer was hung on the wall for readings of room temperature.

The moisture content recorded is the average for the period indicated, as taken from a recording hygrometer, the variations being but slight, as were those of the barometer. All readings were completed within a period of 7.5 hr.

Measurement of Air Flow

The method used for measuring the velocity and quantity of air was based upon the principles of the impact tube and the jet. Briefly, the impact tube, when held in and parallel to the air stream, registers a pressure corresponding to the total energy in the air at that point. In case of continuous flow through a pipe of varying cross-section, if the impact tube is moved up the axis of the air stream, the pressure registered is constant at all points, except for friction losses. The velocity pressure and static pressure vary with every change of cross-section, but the sum of the two, which the impact tube reads, is constant at all points, as the law of conservation of energy indicates. This is similar to Bernouilli's theorem in hydraulics.

Where the section of the pipe is smaller, the velocity of the air must be higher, as the quantity passing all sections of the channel in a given time is constant under conditions of continuous flow. Higher velocity means greater kinetic energy in the moving air particle, and this increment can arise only out of a corresponding diminution of the static pressure.

The jet here used for flow measurements carried this case further, contracting the air column to about one-sixth of its area and discharging into atmosphere at a static pressure equal to atmospheric pressure, or zero U-tube reading, *all* energy in the air being kinetic, read as velocity pressure by the impact tube. This requires that the theoretical orifice be wholly convergent, i.e., that the ratio of absolute pressure of the region into which the jet discharges to the absolute pressure of the region from which the jet discharges be greater than the critical value, 0.5272, for air.

After verifying the fact that throughout the range of velocities used the impact side of the Pitot tube at any given velocity showed constant readings for various positions in the jet, the Pitot was clamped in position, and readings from the impact side only recorded as velocity pressures. At frequent intervals during the runs the static side of the Pitot was tested, but invariably showed zero reading.

The velocity in the jet roughly equaled the velocity through the average valve opening, being about six times the mean velocity in the cylinder proper. In actual magnitude, the velocities ranged from 1500 to 19,000 ft. per min., or 25 to 320 ft. per sec., covering about the extreme range of mean inlet velocities encountered in practice.

Table 1 shows actual and comparative dimensions and areas of the three valve combinations tested.

TABLE 1

Valve Combinations	Circumference of Ports, In.			Cross-Sectional Area of Ports, Sq. In.		
	One Valve	Total	Total in per cent of 2.5-In. Valve	One Valve	Total	Total in per cent of 2.5-In. Valve
2 valves, 1.75-in. diameter.....	5.498	10.996	140	2.405	4.810	97
1 valve, 2.50-in. diameter......	7.854	7.854	100	4.909	4.909	100
2 valves, 1.25-in. diameter.....	3.927	7.854	100	1.227	2.454	50

Diameters and port areas are computed upon the least diameter of the valve or port. In the case of the larger pair, it should be noted that the diameter of 1.75 in., used for convenience, gives an area about 2 per cent less than that required by the geometrical relation for equal area, namely, $D\sqrt{0.5} = 0.7071\,D = 1.768$ in. diameter, for the pair to equal the area of the single valve.

The lifts used with each combination of valves were as follows: 0.05, 0.10, 0.20, 0.30, 0.40, 0.50, 0.75, 1.00 and 1.50 in. These valve lifts were carefully laid off and marked on the stems, and the settings made against fixed indicator points attached to the head of the cylinder. No screw-thread or micrometer arrangement was used, and the probable error was relatively much greater at lower lifts. However, independent settings at low lifts checked within the limit of error of about 2 per cent contemplated for the investigation as a whole. Adjustable clamps were used to hold the valves in position when set, and readings taken covering the pressure range available.

After increasing the lift up to 1.5 in. with each valve combination, the valves were reversed; that is, the stems were clamped in the guides so as to project slightly through the ports, the valve heads remaining entirely outside the cylinder, and readings taken to determine the flow through the ports, eliminating the effect of the valve heads as baffle-plates in the cylinder. It is often stated in works on design that lifting a valve about one-quarter of its diameter develops a valve area equal to that

TABLE 2—CONTINUOUS AIR FLOW THROUGH TWO 1¾-IN. POPPET VALVES

(Humidity, 55 per cent; Barometer, 755 mm)

Velocity Pressure	Square Root Velocity Pressure	Lower Static	Pressure Drop	Square Root Pressure Drop	Upper Static	Temperature, Deg. C		Valve Lift In.	Coefficient Efflux
						Jet	Room		
0.250	0.500	0.20	12.40	3.52	12.75	26.2	23.1	0.05	0.887
0.360	0.600	0.30	16.65	4.09	17.10	26.7	23.1	....	....
0.475	0.689	0.45	22.00	4.69	22.65	29.0	23.3	....	....
0.630	0.794	0.60	28.30	5.33	29.10	31.6	23.5	....	....
0.950	0.975	1.10	11.05	3.33	12.30	27.9	23.7	0.10	0.889
1.520	1.233	1.50	14.90	3.86	16.50	29.1	23.8	....	....
1.710	1.310	1.95	19.25	4.39	21.45	30.8	23.9	....	....
2.850	1.690	2.75	26.95	5.19	29.85	34.3	24.0	....	....
2.680	1.640	2.65	8.45	2.91	11.30	29.1	24.0	0.20	0.818
3.650	1.910	3.50	11.30	3.36	15.00	29.5	..	....	....
4.500	2.120	4.35	13.95	3.74	18.50	30.3	..	....	....
6.350	2.520	6.20	19.15	4.38	25.30	32.5	..	....	....
7.300	2.700	7.10	22.40	4.73	29.65	34.1	24.0	....	....
4.000	2.000	3.80	6.80	2.61	10.85	29.3	24.0	0.30	0.736
5.200	2.280	5.00	8.90	2.98	14.05	29.5	..	....	....
6.350	2.520	6.10	10.75	3.28	16.95	30.0	..	....	....
8.950	2.990	8.60	15.00	3.87	23.65	32.0	..	....	....
11.200	3.450	10.80	18.80	4.33	29.90	34.3	24.0	....	....
4.800	2.190	4.60	5.70	2.38	10.50	29.0	24.0	0.40	0.654
6.150	2.480	5.95	7.40	2.72	14.45	29.5	..	....	....
7.500	2.740	7.30	9.00	3.00	16.40	30.1	..	....	....
10.500	3.240	10.20	12.35	3.51	22.65	31.9	..	....	....
12.750	3.560	12.45	14.90	3.86	27.30	33.4	24.0	....	....
5.500	2.340	5.30	4.95	2.22	10.40	28.9	24.0	0.50	0.608
7.050	2.660	6.90	6.40	2.53	13.45	29.3	..	....	....
8.700	2.950	8.40	7.55	2.75	15.75	30.0	..	....	....
11.750	3.420	11.55	10.35	3.22	22.20	31.6	..	....	....
14.300	3.780	14.15	12.60	3.55	26.90	33.5	24.0	....	....
5.700	2.390	5.50	3.55	1.89	9.30	29.0	24.0	0.75	....
7.450	2.730	7.30	4.65	2.16	12.05	29.0	..	....	0.489
8.900	2.980	8.65	5.40	2.32	14.15	29.5	..	....	....
13.150	3.120	12.95	8.00	2.83	20.95	31.4	..	....	....
16.200	4.020	16.00	9.65	3.11	25.70	33.5	24.0	....	....
6.250	2.500	6.00	3.05	1.75	9.20	28.7	24.0	1.00	0.418
7.800	2.790	7.75	4.00	2.00	11.90	29.0	..	....	....
9.600	3.100	9.25	4.75	2.18	14.10	29.4	..	....	....
14.150	3.760	13.80	6.80	2.61	20.65	31.3	..	....	....
17.200	4.150	16.90	8.30	2.88	25.20	33.0	24.0	....	....
6.400	2.530	6.20	2.80	1.67	9.05	28.1	24.0	1.50	0.296
8.300	2.880	8.05	3.55	1.88	11.70	28.9	..	....	....
9.900	3.140	9.55	4.20	2.05	13.85	29.3	..	....	....
14.500	3.810	14.10	5.85	2.42	20.05	31.0	..	....	....
18.050	4.250	17.50	7.20	2.68	24.90	32.5	24.0	....	....
6.200	2.490	6.05	2.15	1.47	8.40	28.6	24.0	Valve reversed	......
8.200	2.860	8.00	2.80	1.67	10.85	28.8	..		
10.500	3.240	9.65	3.40	1.84	13.15	29.4	..		
14.350	3.780	14.10	4.85	2.20	19.05	31.0	..		
17.550	4.180	17.10	5.90	2.43	23.00	32.7	24.2		

of the port. This is correct if limited to geometric relations, but seriously misleading if interpreted as providing a substantially equal effective orifice, as will later be developed in the experimental results.

Dr. C. E. Lucke has stated that everyone knows that it is of no consequence to lift a poppet valve more than one-quarter of its diameter; that the valve will work better, and the volumetric efficiency and mean effective pressure be better, the larger the diameter of the valve and the smaller the lift.

Experimental Data

Tables 2, 3 and 4 contain the data recorded in the tests of the three valve combinations. They are similar in form and refer, respectively, to the 1.75, 2.5 and 1.25-in. valves. The pressure readings are printed as read, in inches of water.

The readings of velocity pressure in the first column were partly taken on a U-tube inclined at a slope of 10 to 1, to facilitate more accurate readings of small quantities, but the decimal point is recorded so as to show pressures in inches of water, vertical head. Readings taken on the inclined tube are given to three places after the decimal point. After reaching the limit of this inclined tube at about 30 in., or 3 in. actual head, the remaining readings were taken on the usual vertical tubes. This column represents velocity pressure in the jet.

The second column shows the square root of the corresponding reading of velocity pressure in the first column, compared by slide rule. These amounts represent the relative velocities in the jet. The third column or lower static reading refers to the static pressure in the cylinder. When these readings are small the probable error on account of capillarity or inequality in the tubes is rather large, but they were used merely as a rough check on the pressure drop through the valve tested (shown in the fourth column), which was read from a tube connected to both upper and lower statics.

The square root of pressure drop through the valve, computed by slide rule, appears in the fifth column and is proportional to the theoretical mean velocity through the valve. A separate reading on the upper static appears in the sixth column, and the seventh and eighth columns show in degrees centrigrade the considerable variations of temperature with the velocity. The ninth column gives the valve lift, and the tenth column the coefficient of efflux, computed on valve areas equal to πDh, where D = the diameter of the port and h = the valve lift, and assuming that the density of the air was atmospheric.

The conclusion seems to be warranted that the actual velocity through the valve at any given lift varies directly with the square root of the pressure drop, at least within the limits of these tests, as does the theoretical velocity. It further follows as a general rule within these limits that the coefficient of efflux does not vary with the pressure drop.

It is also evident that any projections or sharp angles in the passage tend to greatly reduce the flow, as might

TABLE 3—CONTINUOUS AIR FLOW THROUGH A 2½-IN. POPPET VALVE

(Humidity, 56 per cent; Barometer, 755 mm)

Velocity Pressure	Square Root Velocity Pressure	Lower Static	Pressure Drop	Square Root Pressure Drop	Upper Static	Temperature, Deg. C		Valve Lift In.	Coefficient Efflux
						Jet	Room		
0.140	0.374	0.15	12.60	3.55	12.90	29.0	24.6	0.05	0.960
0.225	0.475	0.25	17.10	4.13	17.40	30.8	..	....	....
0.370	0.608	0.35	23.20	4.82	23.70	32.6	..	....	....
0.435	0.660	0.45	25.75	5.08	26.30	34.3	25.0	....	....
0.505	0.710	0.65	11.35	3.36	12.25	31.2	25.0	0.10	0.950
0.840	0.915	0.90	15.50	3.94	16.60	32.0	..	....	....
1.200	1.097	1.25	20.60	4.55	22.00	33.3	..	....	....
1.655	1.290	1.70	27.75	5.27	29.50	35.6	25.0	....	....
1.690	1.300	1.70	9.55	3.09	11.50	31.4	25.1	0.20	0.856
2.330	1.530	2.35	12.90	3.59	15.40	32.0	..	....	....
3.020	1.740	3.00	16.60	4.07	19.60	33.0	..	....	....
4.000	2.000	3.85	20.85	4.56	25.40	35.2	..	....	....
4.750	2.180	4.50	25.00	5.00	29.65	36.3	25.1	....	....
2.700	1.640	2.55	8.35	2.89	11.10	31.2	25.3	0.30	0.756
3.550	1.880	3.40	11.15	3.34	14.80	31.6	..	....	....
4.550	2.140	4.40	14.30	3.78	18.85	32.4	..	....	....
5.800	2.410	5.70	18.40	4.29	24.20	33.8	..	....	....
7.250	2.690	7.00	22.50	4.75	29.60	35.6	25.3	....	....
3.700	1.920	3.50	6.85	2.62	10.60	30.8	25.6	0.40	0.728
4.850	2.200	4.70	9.15	3.02	13.95	31.2	..	....	....
6.050	2.460	5.90	11.45	3.38	17.40	32.1	..	..	......
7.850	2.800	7.65	14.75	3.84	22.40	33.5	..	....	....
10.200	3.190	9.85	18.65	4.32	28.20	35.8	25.6	....	....
4.400	2.100	4.20	5.60	2.37	10.00	31.3	25.6	0.50	0.710
5.800	2.410	5.60	7.40	2.72	13.20	31.5	..	....	....
7.200	2.680	7.00	9.25	3.04	16.40	32.2	..	....	....
9.550	3.090	9.30	12.10	3.48	21.50	33.6	..	....	....
11.850	3.490	11.45	14.75	3.84	26.00	35.2	25.6	....	....
5.400	2.320	5.20	4.20	2.05	9.60	28.9	25.8	0.75	0.605
7.150	2.670	6.95	5.50	2.34	12.55	30.1	..	....	....
8.900	2.980	8.60	6.80	2.61	15.45	31.4	..	....	....
11.900	3.440	11.55	9.05	3.01	20.70	33.5	..	....	....
14.700	3.880	14.10	10.80	3.28	24.20	35.2	25.8	....	....
5.800	2.410	5.60	3.75	1.94	9.50	31.3	26.3	1.00	0.509
7.700	2.780	7.40	4.90	2.22	12.40	31.8	..	....	....
9.600	3.100	9.25	6.05	2.46	15.40	32.6	..	....	....
12.900	3.590	12.45	7.95	2.82	20.45	34.3	..	....	....
15.200	3.900	14.30	9.05	3.01	22.50	35.3	26.3	....	....
6.100	2.470	5.90	3.30	1.82	9.40	31.8	26.6	1.50	0.373
8.100	2.850	7.90	4.30	2.08	12.25	32.2	..	....	....
10.050	3.180	9.80	5.35	2.32	15.30	33.0	..	....	....
13.500	3.670	13.20	7.05	2.66	20.45	34.5	..	....	....
15.800	3.980	15.00	7.85	2.80	22.20	35.5	26.6	....	....
6.350	2.520	6.15	2.70	1.64	9.05	31.5	26.6	Valve reversed	......
8.500	2.920	8.15	3.50	1.87	11.75	32.0	..		
10.650	3.260	10.30	4.35	2.08	14.75	32.7	..		
14.400	3.800	14.00	5.85	2.42	19.85	34.1	..		
16.700	4.080	15.80	6.40	2.53	21.20	35.3	26.6		

well be anticipated. The very common custom of finishing the inlet passages with two bores meeting at an angle of about 110 deg. certainly puts a heavy restriction upon valve efficiency, but doubtless constructional convenience may be held to justify the practice.

Interesting experimental work could be done on the design of valve guides, possibly joining them to the wall of the passage with a web of streamline section. Valves with an extremely heavy fillet have been used in the R. A. F. 3a engine, doubtless with the idea of guiding the

TABLE 4—CONTINUOUS AIR FLOW THROUGH TWO 1¼-IN. POPPET VALVES

(Humidity, 53 per cent; Barometer, 755 mm)

Velocity Pressure	Square Root Velocity Pressure	Lower Static	Pressure Drop	Square Root Pressure Drop	Upper Static	Temperature, Deg. C		Valve Lift In.	Coefficient Efflux
						Jet	Room		
0.140	0.374	0.20	12.65	3.56	12.90	30.8	27.0	0.05	0.960
0.230	0.484	0.27	17.45	4.18	17.80	32.5	..	....	....
0.350	0.591	0.40	24.10	4.91	24.60	34.5	..	....	....
0.420	0.648	0.45	26.35	5.13	26.85	36.0	27.0	....	....
0.570	0.755	0.65	11.40	3.38	12.05	34.0	27.1	0.10	0.920
0.840	0.917	0.85	16.35	4.04	16.00	34.5	..	....	....
1.120	1.060	1.10	20.10	4.49	21.35	35.5	..	....	....
1.440	1.200	1.40	25.60	5.06	26.10	37.5	27.1	....	....
1.230	1.110	1.20	10.50	3.24	11.85	33.5	27.1	0.20	0.691
1.750	1.320	1.70	14.40	3.79	16.25	34.5	..	..	....
2.330	1.530	2.30	19.25	4.39	21.70	36.0	..	....	....
2.930	1.710	2.95	23.60	4.86	26.60	37.0	27.1	....	....
1.940	1.390	1.90	9.35	3.06	11.30	34.0	27.2	0.30	0.613
2.650	1.630	2.55	12.60	3.55	15.35	34.3	..	....	....
3.550	1.880	3.35	16.50	4.06	20.00	35.0	..	....	....
4.400	2.100	4.20	20.55	4.54	24.75	37.0	..	....	....
5.200	2.280	4.70	24.20	4.92	29.20	39.0	27.2	....	....
2.500	1.580	2.30	8.90	2.98	11.40	34.5	27.2	0.40	0.528
3.350	1.830	3.15	12.00	3.46	15.35	34.7	..	....	....
4.350	2.090	4.10	15.70	3.96	19.90	35.5	..	....	....
5.300	2.300	5.30	20.10	4.49	25.65	38.5	..	....	....
6.250	2.500	6.05	22.20	4.71	28.25	39.0	27.2	....	....
2.700	1.640	2.50	8.50	2.92	11.30	32.0	27.5	0.50	0.459
3.750	1.940	3.55	11.55	3.39	15.10	33.5	..	....	....
4.800	2.190	4.70	15.15	3.89	20.00	35.0	..	....	....
5.850	2.420	5.75	18.40	4.29	24.20	36.5	27.5	....	....
3.100	1.760	2.95	8.35	2.89	11.45	33.5	27.9	0.75	0.327
4.050	2.010	3.90	10.80	3.29	14.80	34.3	..	....	....
5.200	2.280	5.05	14.00	3.74	19.20	35.3	..	....	....
6.600	2.570	6.40	17.70	4.20	24.05	37.0	27.9	....	....
3.100	1.760	2.95	7.80	2.79	10.90	34.0	27.9	1.00	0.254
4.200	2.050	4.00	10.45	3.24	14.60	34.6	..	....	....
5.750	2.400	5.60	14.25	3.78	20.00	35.5	..	....	....
7.100	2.660	6.90	17.65	4.20	24.60	37.3	..	....	....
3.350	1.830	3.20	7.90	2.81	11.25	34.0	27.9	1.50	0.174
4.350	2.080	4.10	10.25	3.20	14.45	34.3	..	....	....
6.000	2.450	5.85	13.60	3.68	19.30	35.6	..	....	....
7.100	2.660	6.90	16.50	4.06	23.50	37.0	..	....	....
3.350	1.830	3.15	7.50	2.74	10.90	33.5	27.9	Valve reversed	......
4.600	2.150	4.30	10.05	3.17	14.60	34.2	..		
5.900	2.430	5.75	13.25	3.64	19.10	35.0	..		
7.250	2.690	7.10	16.20	4.02	23.35	36.5	..		

air current smoothly to the valve opening. Venturi effects in the passage immediately above the valve might be productive of excellent results.

If flow curves are plotted in terms of the proportional flow and the valve lift for these valve combinations, it is possible to compare the vertical intercepts of these curves on ordinates corresponding to various valve lifts in percentages of the flow of the single valve, as in Table 5.

The points connected to the curves by broken lines indicate, to vertical scale only, the flow with the valves reversed. This might be considered equivalent to the flow with the valves at an infinite lift, which agrees with the horizontal trend of the curves, but more practically, these points represent the maximum limit of flow through the respective ports at this pressure drop. These curves, comparing performance upon a basis of equal lift in inches, are particularly applicable where it is conceded that mechanical features generally limit the possible lift regardless of valve diameter.

On the other hand, it is often asserted that the proper limit of valve lift is a function of the diameter, and for

TABLE 5

Lift, in.	0.125	0.250	0.375	0.500	0.625	0.750
2 valves, 1.75-in. diameter	137%	129%	125%	119%	116%	113%
*2 valves, 1.50-in. diameter	114%	106%	101%	94%	90%	87%
1 valve, 2.50-in. diameter	100%	100%	100%	100%	100%	100%
2 valves, 1.25-in. diameter	85%	77%	72%	65%	59%	54%

*Interpolated.

purposes of comparison on this basis, Fig. 3 has been prepared from the curves of Fig. 2, changing the horizontal scale to read in per cent of the diameter of each valve. In the case of pairs of valves, the flow of both is plotted against the lift, expressed in per cent of the diameter of one valve only.

The result of this transposition is at once apparent. The intercepts on any ordinate very closely agree with the proportionate cross-sectional port areas of the several valve combinations, and in the case of the two curves corresponding to valve combinations with equal cross-sectional port area, the curves coincide within the probable error of the work. Up to a lift of 0.5 diameter the coincidence is all the more exact if it be remembered that the two 1.75-in. valves have an area about 2 per cent less than the single 2.5-in. valve.

From this it would appear reasonable to infer that under fairly similar conditions different valves or combinations of valves have capacities in proportion to their respective cross-sectional port areas, when the lift in each case is the same per cent of their respective diameters.

As a rough comparison of probable friction loss and dynamic loss, it may be assumed that the friction in the passage and at the lip of the valve is equivalent to that of five diameters of straight pipe at the same velocity. Dynamic losses might be expected equal to the friction loss in about eight diameters due to the curvature of the passage, and further dynamic losses equal to the friction in at least thirty diameters due to the sharp change of direction at the valve seat, -60 deg. at low lift with a 30-deg. seat. If this comparison is within the limits of fair approximation, Mr. Pomeroy's 39 per cent greater friction loss is applicable to about 15 per cent only of the total loss, or about 6 per cent less capacity would be expected from a pair of valves having 0.7 the diameter and 0.7 the lift of a single valve. From the data here obtained it appears that the two valves have only about 2 per cent less capacity at 0.7 the lift.

For ready comparison, Table 6 has been prepared from the vertical intercepts on the curves of Fig. 3, showing the relative cross-sectional port areas and capacities in per cent of the area and capacity of the single 2.5-in. valve, each valve being lifted the same per cent of its diameter.

It is evident from an inspection of the curves in Figs.

TABLE 6

	Relative Area	Relative Flow at Lift Equal to—			
		0.10 Diameter	0.15 Diameter	0.20 Diameter	0.25 Diameter
2 valves, 1.75-in. diameter	98%	96%	96%	96%	96%
2 valves, 1.50-in. diameter*	72%	70%	69%	69%	69%
1 valve, 2.50-in. diameter	100%	100%	100%	100%	100%
2 valves, 1.25-in. diameter	50%	46%	45%	45%	45%

*By interpolation.

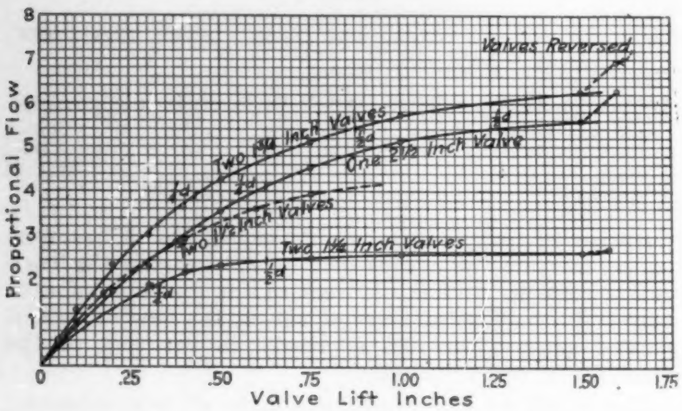

FIG. 2—PROPORTIONAL FLOW OF VARIOUS VALVE COMBINATIONS FOR DIFFERENT VALVE LIFTS IN INCHES

2 and 3 that a lift equal to one-quarter diameter develops less than 67 per cent of the full capacity of the port, and that a lift of one-half diameter develops 80 to 90 per cent of the full capacity.

The coefficient of efflux is taken as the ratio of the observed mean velocity through the valve to the mean velocity which would theoretically result from an equal pressure drop. Assuming that the temperature, density and humidity of the air are the same at the valve as at the jet, this coefficient may be obtained directly from the relation of the areas and the proportional velocities set forth in Tables 2, 3 and 4. The proportional velocity at the jet multiplied by the ratio of the jet area to valve area gives the proportional velocity through the valve. If the ratio of this velocity to the square root of the pressure drop be taken, the result is the coefficient of efflux. To be more exact, this should be multiplied by 0.99, the coefficient of the jet.

In computing the coefficient the valve area has been taken as πDh for all lifts. It is realized that for small lifts the aid of trigonometrical formulas may be invoked to determine accurately the least area of opening, but the same formulas are not applicable at higher lifts. Moreover, they are only justifiable upon the theory that the lines of flow are parallel to the slope of the valve seat, a condition which certainly does not obtain for any except the smallest lifts.

In Fig. 4 the coefficients of efflux will be found, plotted against valve lift in inches at the bottom, and against lift in per cent of diameter above. These coefficients are considerably higher at low lifts, a feature somewhat difficult to explain satisfactorily. Both friction and dynamic losses should be greater at low lifts, as the ratio of perimeter to area is then greater and the angular deflection sharper. It seems probable that there is an approximation to a jet action at low lifts, the discharge taking place into a region of relatively low pressure, somewhat after the manner of the true jet used for measurement at the outlet end of the cylinder. The comparatively high discharge efficiency of any such jet seems to make this the most probable explanation of the high coefficients.

If such jet action takes place, the pressure in the valve area should approximate that of the cylinder itself, and the theoretical velocity through the valve should be computed upon the lower pressure rather than the higher. This would reduce the error involved in computing the theoretical flow and coefficient upon the assumption of atmospheric density in the valve, as has been done.

The maximum static pressure in the cylinder was 17.5 in. of water. As a pressure of 1 in. of water is equal to a pressure of 0.5768 oz. per sq. in., this would equal a pressure of 10.09 oz., or 0.631 lb. per sq. in., or an absolute pressure of 15.23 lb. per sq. in., 755 mm. observed atmospheric pressure being equal to 14.60 lb. per sq. in. The density of the air varying with the absolute pressure and the ratio of absolute pressures being 1.046, the error involved under the above assumptions would be about 2.2 per cent as the square root of the density of the air enters into the calculations. This error would be materially less at the lower lifts, the pressures in the cylinder then being considerably less. No appreciable error would appear to be introduced by assuming equal temperature and equal humidity at valve and jet for any given valve opening and pressure drop.

Referring again to Fig. 4, it will be noted that in the lower set of curves, where the coefficients are compared at the same absolute lift, the differences between the three valve combinations are considerable, and that at the very low lifts the points plotted present some irregularities. The curves have been drawn to conform to the greatest number of points reasonably possible, and the curves in upper portion have been plotted from them. The points for the two larger combinations so nearly coincide in the upper set that but one line has been drawn.

The relative intercepts of the coefficient curves at various absolute lifts, expressed in per cent of the values for the single 2.5-in. valve, are presented in Table 7.

TABLE 7

	Relative Coefficient of Efflux				
Valve lift, in.	0.125	0.25	0.375	0.50	0.625
2 valves, 1.75-in. diameter	96%	94%	91%	89%	86%
1 valve, 2.50-in. diameter	100%	100%	100%	100%	100%
2 valves, 1.25-in. diameter	87%	79%	73%	69%	64%

In the upper set of curves of Fig. 4 it will be seen that when compared on a basis of equal valve lifts, expressed in per cent of diameter, the coefficients are much more nearly equal, the curves for the two larger combinations coinciding, and that for the small valves being but little lower. It seems entirely probable that even this small difference is caused largely by the converging lines of the

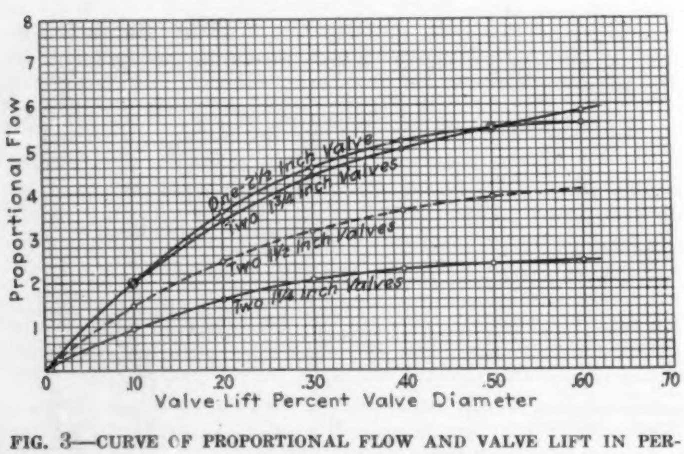

FIG. 3—CURVE OF PROPORTIONAL FLOW AND VALVE LIFT IN PERCENTAGE OF VALVE DIAMETER

passages leading to these small valves, as before explained. The comparative values are given in Table 8.

TABLE 8

	Relative Coefficient of Efflux				
Valve lift in per cent of diameter....................	0.10	0.15	0.20	0.25	0.30
2 valves, 1.75-in. diameter...	100%	100%	100%	100%	100%
1 valve, 2.50-in. diameter...	100%	100%	100%	100%	100%
2 valves, 1.25-in. diameter...	93%	92%	91%	90%	90%

General Considerations

It is patent that extreme care should be exercised in any attempt to apply the results of continuous-flow experiments to flow under operating or intermittent conditions, since inertia and resonance effects in the inlet manifold will obviously make great differences in the absolute quantities, and these effects will vary with the type of manifold used. Moreover, the pressure drop, velocity and coefficient will obviously vary with many other factors as between different engines, different speeds for the same engine, and as to instantaneous values at different points of the stroke for a given engine at a given speed.

However, in the question of design as to whether two inlet valves or one should be used, it is believed the comparative results here presented may be made to serve a real purpose. It is difficult to perceive any reason why the comparative relations obtaining between these three valve combinations for continuous flow should not find some parallel in the comparative relations between the same three combinations for intermittent flow, if no other variables are permitted to affect the comparative results in the latter case. Only inherent differences between the three combinations, effective with intermittent flow and non-effective with continuous flow, or vice versa, would appear capable of affecting this parallel, and it is improbable that such differences, if any, are of great magnitude.

The dimensions of the cylinder model used for these experiments offer a ready basis for discussion, and are commonly encountered in aviation engine practice, the bore being 5 in. and the diameter of combustion chamber 5.75 in. A combustion chamber of this size permits the use of two valves of 2.5-in. diameter, or four valves of 1.875-in. diameter, inclined at 15 or 20 deg. to the cylinder axis in both cases. Four 1.75-in. valves can be placed in a 5.5-in. inclined cylinder head, or a 5.75-in. vertical cylinder head, and four 1.5-in. valves are even more readily accommodated in a 5-in. cylinder head, or a cylinder having the combustion chamber the same diameter as the cylinder proper. These valves may be placed vertically, and the cylinder is much more easily machined. The combustion chamber will have better proportions, and the slight increase in cylinder height will be more than offset as to over-all height by the saving in spring length.

Two 1.5-in. valves will have a flow capacity equal to one 2.5-in. valve at the same pressure drop and the same lift, will present but 72 per cent as much area to any pressure in the cylinder at the time of opening, and will have but 56 per cent of the weight of the single valve, assuming that the weights vary at $D^{2.5}$, which is approximately correct for these sizes. Assuming any reasonable pressure in the cylinder at the time of valve opening, and spring tensions in proportion to valve weights, it is evident that the two small valves will require less than half the power to open them, and this will be a direct saving of mechanical loss, as valve action is not the type of reciprocating motion which can return during one portion of the stroke energy stored during another portion, excepting only the energy stored in the spring.

It has been said that valves in pairs are more difficult to cool than single valves, but this does not appear to stand anaylsis. The proportion of the 5-in. cylinder head occupied by the small valves is only about 95 per cent of the proportion of the 5.75-in. head occupied by the large valve. The circumference of the two valves is 20 per cent greater than that of the single valve, and although the seats would have somewhat less width, the distance of heat flow in this direction would be but 60 per cent as great. As to the portion of the heat which flows to the

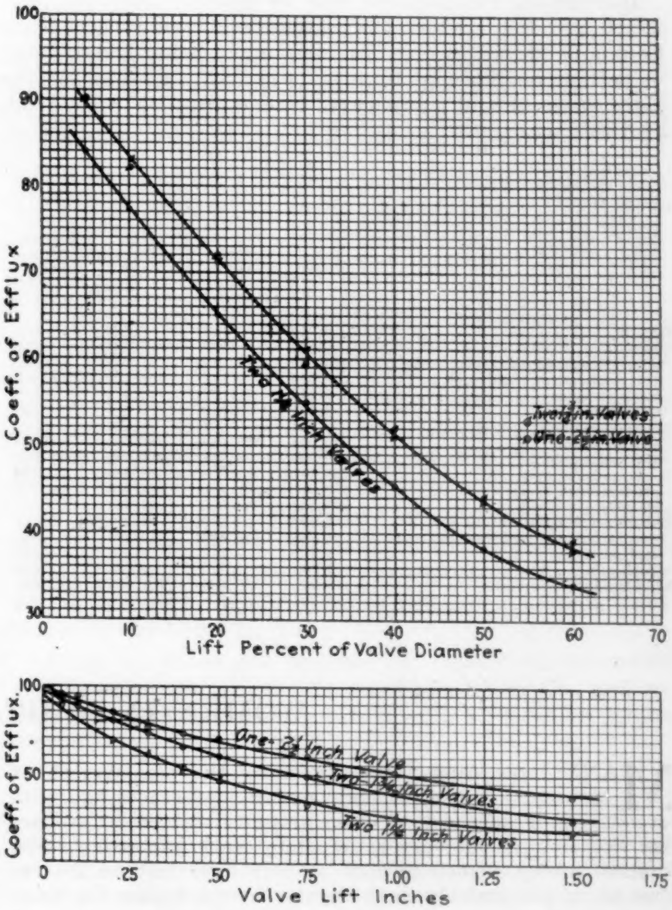

FIG. 4—CURVES OF THE COEFFICIENT OF EFFLUX FOR VARIOUS VALVE COMBINATIONS PLOTTED AGAINST VALVE LIFT IN INCHES AND AS A PERCENTAGE OF THE DIAMETER

guide, the conditions are also somewhat in favor of the small valves, the distance to the water-cooled portion of the guide being less and the proportion of water-cooled guide greater.

In one example of foreign engine design dual valves of about this size are lifted to one-half diameter, and give entirely satisfactory operation at speeds up to 2200 r.p.m. The possibilities in this direction are largely untried, but the negative work used in overcoming valve resistance to inlet flow might be reduced with small valves

at high lifts and the volumetric efficiency increased, without introducing serious mechanical difficulties. This, of course, is contrary to the principle of using low lifts to secure a higher coefficient, but still the over-all result might be beneficial.

The comparison of a single 2.5-in. valve with a pair of 1.75-in. valves may be analyzed in much the same manner, and as to heat conditions the result would seem slightly in favor of the pair. If lifted 0.375 in., the capacity will be 25 per cent greater than that of one 2.5-in. valve, according to the experimental results shown in Table 5, or the resistance will be but 64 per cent as great, the resistance varying approximately with the square of velocity or capacity. This should result in higher volumetric efficiency. The superficial area of the two combinations would be practically equal, but the weight of the pair would be but 82 per cent of that of the single valve, with correspondingly reduced total spring tension and slightly reduced mechanical loss.

Interesting comparisons may be drawn from data published by the *Automobile Engineer*, London, covering Benz and Mercedes engines, each make being constructed in both two-valve and four-valve models. Except for the valve changes and an increase in compression ratio from 4.50 to about 4.90, the design of the four-valve models is much the same as that of the respective two-valve types. The data are represented in Table 9, the ratio of volume to horsepower and brake mean pressure being given for the rated power at 1400 r.p.m. for each engine. The "valve factor" is one-half the product of inlet-valve opening area by the number of degrees open divided by the displacement of one piston, affording a ready index of relative valve capacity.

TABLE 9

Engine	Benz Four-Valve	Mercedes Four-Valve	Benz Two-Valve	Mercedes Two-Valve
Bore, in	5.710	6.300	5.120	5.510
Stroke, in	7.480	7.090	7.090	6.300
Piston displacement per cylinder, cu. in.	191.380	220.820	146.050	150.200
Piston displacement, total, cu. in	1,148.300	1,324.900	876.300	901.200
Valve port diameter, in	2.040	2.170	2.420	2.670
Valve lift, in	0.465	0.398	0.433	0.440
Inlet valve opening, sq. in	2.990	2.720	3.290	3.700
Rated horsepower	230.000	260.000	160.000	160.000
Rated speed, r.p.m	1,400.000	1,400.000	1,400.000	1,400.000
Maximum horsepower	250.000	270.000	164.000	162.500
Maximum speed, r.p.m	1,650.000	1,650.000	1,400.000	1,400.000
Inlet-valve opening, deg	245.000	228.300	240.000	213.000
Area inlet pipe, sq. in	3.650	6.850	2.960	3.540
Piston displacement per horsepower, cu. in	4.990	5.100	5.480	5.630
Compression ratio	4.910	4.940	4.500	4.500
Valve factor	3.800	2.820	2.700	2.620
Brake mean effective pressure	113.000	107.500	103.000	102.000

The valve factor for the four-valve Mercedes is but slightly larger than that of the two-valve, and the mean effective pressure is increased only 5 per cent, which is practically accounted for by the increase in compression ratio from 4.50 to 4.94. In the Benz four-valve, the factor is increased 35 per cent and the mean effective pressure increased 10 per cent, only about one-half of which can be due to the increase in compression ratio from 4.50 to 4.91.

CONCLUSIONS

The conclusions drawn by the investigators are as follows:

(1) The coefficient of efflux is practically constant, for all pressure drops (at least below 1 lb. per sq. in.) where the lower pressure is approximately atmospheric, and the theoretical flow is computed upon air at atmospheric density.

(2) Under conditions of general similarity, the coefficient of efflux is very nearly the same for valves of different sizes, at equal lifts expressed in per cent of their respective diameters.

(3) Lifting a valve one-quarter of its diameter may develop an area of opening geometrically equal to its port area, but affords a capacity less than 67 per cent that of the unobstructed port, at the same pressure drop; a lift equal to one-half diameter develops 80 to 90 per cent of this maximum capacity.

(4) At the same pressure drop, one valve of diameter D and lift h is equal in capacity to a pair of valves of diameter 0.707 D, equal port area, and lift 0.707 h, or a pair of valves of diameter 0.6 D and lift h, for values of h not exceeding about 0.25 D.

WOMEN IN AIRPLANE PRODUCTION

MANY interesting facts regarding the employment of women have been brought out in conferences had with foremen. It is agreed that when properly trained on production women turn out very satisfactory work. In many cases foremen prefer women to men, as they have learned that in some work, particularly on the lighter forms, women are more adaptable than men. Women can do doping, painting and varnishing equal to men previously employed on such operations. In some branches of woodwork foremen report that a slightly greater number of women are required to secure the same amount of production formerly obtained from men who were experienced mechanics in woodworking lines. However, as women acquire experience their quantity of production increases rapidly. From the beginning the quality has been superior to that previously obtained from the male employes whose chief aim was quantity of production.

The superintendent of the wire department is greatly pleased with the results obtained from the women under him. His experience confirms the statement already made that the quality of the women's work is higher than that formerly obtained from men, and that women very rapidly equal the quantity of production.

Where women have been introduced into mechanical activities without training both employes and foremen have become discouraged. This condition does not prevail where suitable training is first given to the women, as instruction insures their adaptation to the work for which they are best fitted, and at the same time provides for the foremen help that is ready to produce without further instruction.

—F. L. Colynn

PERSONAL NOTES OF THE MEMBERS

Harold L. Arnold is in Los Angeles, Cal., acting as service distributor of the National Carbon Co.

G. Edward Barnhart has accepted the position of chief engineer in the Handley-Page engineering department of the Standard Aircraft Corporation, Elizabeth, N. J.

P. J. F. Batenburg has resigned his position as chief engineer of the Four Wheel Drive Auto Co., Clintonville, Wis. He has been associated with this company for the past eight years and designed the four-wheel drive on which the rapid success of the company was founded. He has as yet made no business connection.

Frank E. Blanchard, until recently with the Fisher Co., has returned to his former position as engineer of the motor truck division of the Milburn Wagon Co., Toledo, Ohio.

C. H. Dunlap has terminated his supervision of Motor Transport Corps work at the Students' Army Training Corps Detachment, Valparaiso, Ind., and returned to his duties as vice-president and sales manager of the E. A. Nelson Motor Car Co., Detroit.

E. S. Echlin, recently discharged from army service, has returned to his old position with the J. I. Case Threshing Machine Co., Racine, Wis.

Thomas T. Fauntleroy, manager central sales district, Gurney Ball Bearing Co., with headquarters at Cleveland, was appointed sales engineer of the Lakewood Engineering Co., Lakewood, Cleveland, on Jan. 1. He has been with the Gurney company for the past three and a half years, being sales engineer for the Eastern office until April, 1918, when he was transferred to Cleveland.

H. J. Garceau has been appointed sales manager of the Warner Gear Co., Muncie, Ind.

G. Walker Gilmer, Jr., formerly purchasing agent and sales manager of the L. H. Gilmer Co., Tacony, Philadelphia, Pa., has been made chief of engineering of the same organization.

E. R. Godfrey, until recently chief draftsman for the Remy Electric Division, United Motors Corporation, Anderson, Ind., is now assistant to the vice-president, Midwest Engine Co., Indianapolis, Ind.

Edward C. Hach has been discharged from army service and is for the present at his home in East Lansing, Mich.

Harlie H. Hicks has returned from service overseas and is now at his home in Orion, Mich.

E. K. Hill, formerly with the inspection department of the Wright-Martin Aircraft Corporation, New Brunswick, N. J., has accepted the position of mechanical draftsman for the E. J. Longyear Co., exploring engineers, Minneapolis, Minn., and builders of drills and other mining machinery.

Jacques de Jong, in service with the Belgian troops until the signing of the armistice, will take his old position with the Minerva Motors S. A. in Antwerp.

C. S. Kegerreis has received his discharge from the army and is again doing research work on carburetion at Purdue University, Lafayette, Ind.

George E. McGill has been honorably discharged from the army and is back in his former position as draftsman for the Lincoln Motor Co., Detroit, Mich.

E. E. Minard, who was formerly with the J. R. Stone Tool & Supply Co., Detroit, Mich., as engineer and salesman, has been appointed president and general manager of the National Sales Engineering Corporation of that city.

S. B. Roberts has accepted a position with Bartram Brothers, Inc., Santo Domingo City, West Indies. He will act as engineer on a large sugar plantation.

O. J. Rohde, formerly manager of the service department of the Wire Wheel Corporation of America, Buffalo, N. Y., with headquarters in New York City, has been appointed vice-president and general manager of the corporation.

N. G. Rost, general sales manager, Duesenberg Motors Corporation, has returned from a trip of three months to France and England in the interest of the corporation and is now located at 120 Broadway, New York City.

Le Roy Scott, formerly engaged in sales promotion work for the storage battery division of the National Carbon Co., Cleveland, Ohio, is now acting as consulting engineer for the storage battery department of the company's San Francisco office and as manager of the electrical and battery department of the company in Los Angeles, Cal.

Vernon G. Souder, who has recently been connected with the Washington office of the General Motors Corporation as service engineer, has returned to the New York office of the General Motors Export Co., where he was prior to his transfer to Washington.

Malcolm Thomson has been honorably discharged from military service in the Altitude Laboratory of the Bureau of Standards' Aircraft Production Department at Washington, and is for the present at his home in Swampscott, Mass.

J. G. Vincent, vice-president of engineering in the Packard Motor Car Co., has been released from Government service and is back at his post in Detroit.

R. A. Watkins, until recently with the Bureau of Steam Engineering, Navy Department, Washington, is now vice-president and general manager of the Bath Machine Works, Inc., Bath, N. Y.

R. E. Wildrig, released from the army, is now with the Garford Motor Truck Co., Lima, Ohio.

W. H. Wilson has left the Lincoln Motor Co., Detroit, and is now with the Wright-Martin Aircraft Corporation, Long Island City, N. Y.

Frank R. Wood has severed his connection as service manager for the Lippard-Stewart Motor Car Co., and will act in the same capacity for the H. J. Koehler Motors Corporation, Buffalo, N. Y.

ACTIVITIES OF S. A. E. SECTIONS

THE January meeting of the Mid-West Section was held at the Western Society of Engineers, Chicago, on the 10th. A. H. Wyatt, president and general manager of the Automotive Corporation, Fort Wayne, Ind., presented a paper on the one-man tractor.

An address was delivered by Major Frank B. Gilbreth, consulting engineer, on "Modern Methods of Transferring Skill", at the January meeting of the Cleveland Section. This was a joint meeting with the Cleveland Engineering Society and was held on the 14th. A very elaborate set of motion pictures supplemented the address.

The January meeting of the Detroit Section was held on the 24th. An address on "Patents and Inventions and How to Handle Them", was delivered by B. M. Kent, patent counsel of the Standard Parts Co.

The meeting of the Buffalo Section scheduled for Feb. 5 has been cancelled on account of the conflict with the date of the annual meeting of the Society.

OBITUARIES

H. W. Ford, who was probably best known in the motor car industry as the president of the Saxon Motor Car Corporation, died in New York on Nov. 18 of pneumonia, aged 38 years. He was born at Knob Noster, Mo., May 4, 1880, and received his early education in the schools of his native state. His education was completed in the high school at Chicago and the University of Chicago. In 1910 Mr. Ford went to Detroit from Dayton, Ohio, and entered the employ of the Chalmers Motor Car Co., becoming its secretary and advertising manager. In 1914, with a number of other Chalmers employes, he founded the Saxon Motor Car Corporation and was elected president. He retired from this position last January to become president of the Federal Bond & Mortgage Co. He entered the service of the Motor Transport Corps and was stationed at Camp Gordon, Ga., being recently discharged with the rank of captain. He was elected to the Associate Member grade of the Society in January, 1914. His widow and two daughters survive.

Lieut. George R. Mason died of bronchial pneumonia in France, Sept. 9, 1918, aged 29 years. He was born at Des Moines, Iowa, in 1890. In the schools his most successful work was done in mathematics and science, while his leisure was passed in automobile driving. At seventeen he was doing contest driving and followed this four years, picking up on the road and in garages the varied experience, ingenuity and quick decision which advanced him later on through the drafting and designing rooms to the supervision of all repair and adjustment work in the Mason Motor Co., Waterloo, Iowa. He was assistant engineer when he enlisted and was among the first of the Waterloo, Iowa, men to cross to France with the American Expeditionary Force. Lieut. Mason had been a Member of the Society since 1912.

Robert Paul Patterson, a Member of the Society, died Dec. 19, at his home in Luck, Wis. He was born at Milltown, Wis., June 14, 1895, and went to Luck in 1901 with his parents. After graduating from the elementary schools of Luck, he attended the St. Croix Falls high school and was graduated in 1912. He completed the course in mechanical engineering at the University of Wisconsin in 1916 and entered the employ of the Curtiss Aeroplane & Motor Corporation as assistant engineer, where he specialized in the adaptation of gas engines for airplane use. At one time Mr. Patterson had charge of the production department of the Emerson-Brantingham Co. at Minneapolis, and later was in charge of the distributing and retailing end of the Elcar Motor Sales Co., at Luck, although his specialty was automobile and gas engines. He also did some work in the construction of high-tension electrical transmission lines. Mr. Patterson was elected to membership in the Society in June, 1917, and was an active member of the Aero Section of the S. A. E. Committee to prepare standards for aircraft parts.

John F. Reno died of pneumonia Dec. 19, at his home in East Moline, Ill. He was born in Illinois in 1883, and took the course in mechanical engineering at the University of Illinois, graduating with the class of 1908. After serving for a short interval as chief draftsman for the Root & Van Dervoort Engineering Co. he engaged with the Moline Automobile Co. and remained with this organization until his death. He designed the engines manufactured by this company, developed their chassis and found time for much experimental work. He was especially interested in the Knight sleeve-valve and in poppet-valve engines and gave a great deal of time to their adaptation and successful working. Mr. Reno was elected to membership in the Society in June, 1917.

Applicants *for* Membership

The applications for membership received between Dec. 15, 1918, and Jan. 20, 1919, are given below. The members of the Society are urged to send any pertinent information with regard to these names which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

ALLING, CLAUDE R., engineer, casualty department, Underwriters' Laboratories, *Chicago, Ill.*

ARMSTRONG, RALPH W., designer, Edward V. Hartford, Inc., *Jersey City, N. J.*

ARNOLD, ALBERT, foreman, Gray Motor Co., *Detroit, Mich.*

BALDWIN, LEONARD C., draftsman, H. J. Koehler Motors Corporation, *Newark, N. J.*

DE BELLEUSE, ALBERT C., production expert, Aircraft Production Bureau, Finance Department, *New York City.*

BLAIR, WILLIAM RICHARD, works manager, National Tool & Mfg. Co., *St. Louis, Mo.*

BODENWASSER, ELIAS, JR., sergeant, Motor Overhaul Shop, A. P. O., 717, A. E. F., *France.*

CARLSON, WILLIAM S., mechanical engineer, Byrne Kingston & Co., *Kokomo, Ind.*

CHANDLER, FRANKLIN F., manager of sales, Chandler & Taylor Co., *Indianapolis, Ind.*

CLARK, SAMUEL, manager, Ellwood Ivins Tube Works, Oak Lane, *Philadelphia, Pa.*

COCHRAN, BURL S., production engineer, Grant Lees Gear Co., *Cleveland, Ohio.*

CHRONIC, CLARENCE, tractor designer, J. I. Case Threshing Machine Co., *Racine, Wis.*

CULVER, EDWARD P., second-lieutenant engineer, Air Service, *Hudson Falls, N. Y.*

DORSEY, S. H., chief engineer, Traffic Motor Truck Corporation, *St. Louis, Mo.*

EDGERTON, C. W., acting service manager, Wright-Martin Aircraft Corporation, *New Brunswick, N. J.*

EWING, JOSEPH, sales manager, Haskelite Mfg. Co., *Grand Rapids, Mich.*

FARNSWORTH, THOMAS WEBSTER, mechanical engineer, 346 Fairfield Ave., *Hartford, Conn.*

FRANKEL, MORTIMER, assistant sales manager, Roller Smith Co., *New York City.*

GROSELLE, JOHN FRANCIS, checker of mechanical drafting, Peerless Motor Car Co., *Cleveland, Ohio.*

HARPER, FRED. CLAYTON, proprietor, F. C. Harper Screw Works, *Chicago, Ill.*

HAZARD, GEORGE F., mechanical engineer, Kellogg Manufacturing Co., *Rochester, N. Y.*

HOLCOMB, J. ROGERS, president, Electric Steel Co., *Indianapolis, Ind.*

HOLLINGER, HAROLD D., superintendent, Wright-Martin Aircraft Corporation, *New Brunswick, N. J.*

HUTCHINS, W. H., engineering department, North East Electric Co., *Rochester, N. Y.*

IRELAND, W. S., general manager, National Tool & Mfg. Co., *St. Louis, Mo.*

IRWIN, GEORGE A., lieutenant, Motor Transport Corps, *Washington.*

JENKINS, A. F., general superintendent, Dort Motor Car Co., *Flint, Mich.*

JONES, HUGH IVOR, assistant metallurgist, Willys-Morrow Co., *Elmira, N. Y.*

KINSEY, OWEN D., general storekeeper and head of planning dept., Buda Co., *Harvey, Ill.*

KLECKLER, HARRY, superintendent, Curtiss Aeroplane & Motor Corporation, *Buffalo, N. Y.*

LA MANNA, THOMAS G., chief draftsman, Pomilio Brothers Corporation, Aviation Experimental Works, Speedway, *Indianapolis, Ind.*

LEDWINKA, JOSEPH, chief engineer, Edward G. Budd Mfg. Co., *Philadelphia Pa.*

LJUNGLOF, GEORGE LUDWIG, draftsman, Engineering Division, Motor Transport Corps, *Washington*.

LOOP, J. F., chief engineer, Graham Brothers, *Evansville, Ind.*

MCMULLEN, ALBERT T., draftsman and designer, Lincoln Motor Co., *Detroit, Mich.*

MACK, ERNEST D., secretary and treasurer, Mack Bros., Inc., *Reno, Nev.*

MALM, WALTER R., district metallurgist, Bureau of Aircraft Production, *San Francisco, Cal.*

MEYER, WILLIAM H., chief draftsman, National Tool & Mfg. Co., *St. Louis, Mo.*

MILLER, WESLEY, senior inspector, Buffalo District, Bureau of Aircraft Production, *Buffalo, N. Y.*

MITCHELL, F. ROBINS, president and treasurer, Mitchell & Smith, Inc., *Boston, Mass.*

MORGAN, HARRY, chief engineer, Walker Weiss Axle Co., *Flint, Mich.*

MORROW, J. G., inspecting engineer and metallurgist, Steel Company of Canada, *Hamilton, Ont.*

MOSES, EDMUND Q., patent lawyer, 52 Broadway, *New York City.*

NESS, NORMAN R., draftsman, motor engine department, Minneapolis Steel Machinery Co., *Minneapolis, Minn.*

[illegible] ERIK, editor in chief, *Machinery, New York City.*

RICHIE, GEORGE ELLIOTT, JR., superintendent of repairs, Autocar Sales & Service Co., *Chicago, Ill.*

RISBERG, M. M., comptroller, Republic Motor Truck Co., Alma, Mich.; comptroller, Torbensen Axle Co., Cleveland, Ohio; comptroller, M & S Corporation, *Detroit, Mich.*

ROSS, EDWIN A., inspector, Navy Aircraft Bureau of Construction & Repair, factory of Curtiss Aeroplane & Motor Corporation, *Buffalo, N. Y.*

SCHMITZ, WILLIAM JOHN, mechanical draftsman, Ordnance Experimental Station, *Speedway City, Ind.*

SCHOETTLER, IRWIN, tool designer, Nordyke & Marmon Co., *Indianapolis, Ind.*

SCOTT, LEWIS L., chief engineer, Standard Engineering Co., *St. Louis, Mo.*

SIEBER, C. A., draftsman, Ordnance Department, Motor Equipment Section, *Washington.*

SMITH, FREDERICK R., chief designer, Siddeley Deasy Motor Car Co., Ltd., *Coventry, England.*

SMITH, HAROLD HOOPER, electrical engineer, Edison Storage Battery Co., *Orange, N. J.*

TURNER, WILLIAM S., senior inspector, airplanes and airplane engines, Bureau of Aircraft Production, *Detroit, Mich.*

VAN VLIET, JOHN D., aeronautical engineer, Standard Aircraft Corporation, *Plainfield, N. J.*

WERTZHEISER, JOSEPH, chief draftsman, Lawrence Aero Engine Co., *New York City.*

WILLETTS, ELWOOD H., first-lieutenant, Ordnance Reserve Corps, Motor Equipment Section, *Clintonville, Wis.*

WILLIAMS, FRED. D., general manager, L. H. Gilmer Co., *Philadelphia, Pa.*

WRIGHT, JAMES A., assistant section head, Bureau of Aircraft Production, *Detroit, Mich.*

ZURLINDEN, WILLIAM W., operator, Utilities, Quartermaster Corps, *Camp Dodge, Iowa.*

Applicants *Qualified*

The following applicants have qualified for admission to the Society between Dec. 15, 1918, and Jan. 16, 1919. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff.) Affiliate; (Aff. Rep.) Affiliate Representative; (S. E.) Student Enrollment.

ACTON, MICHAEL J. (M) superintendent, works manager, Lapointe Machine Tool Co., *Hudson, Mass.* (mail) 75 Maple Street.

ANDERSON, BIRGER N. (J) mechanical draftsman, Four Wheel Drive Auto Co., *Clintonville, Wis.*

AUTENRIETH, GEORGE C. (M) assistant professor in charge of Gas Power Laboratories, College of the City of New York, St. Nicholas Terrace and 139th Street, *New York City.*

BATCHELDER, CHARLES F. (A) Olds Motor Works, *Lansing, Mich.*

BECK, HAROLD MACKNIGHT (M) construction and operating department engineer, Electric Storage Battery Co., Nineteenth Street and Allegheny Avenue, Philadelphia, Pa. (mail) 1730 Marquette Building, *Chicago, Ill.*

BEUTEL, ROBERT J. (M) designer, International Motor Co., Sixty-fourth Street and West End Avenue, New York City (mail) 120 Crystal Street, *Brooklyn, N. Y.*

BOBSENE, FRED. J. (A) superintendent of assembly, Curtiss Aeroplane & Motor Corporation, 2000 North Elmwood Avenue, *Buffalo, N. Y.* (mail) 926½ Main St.

BRADLEY, S. L. (A) sales manager, Ross Gear & Tool Co., *Lafayette, Ind.*

BUERKLE, LEWIS J. (A) electrical engineer, Dayton Engineering Laboratories Co., *Dayton, Ohio,* (mail) 28 Webb Avenue.

BURKE, HAROLD J. (J) inspector airplanes and engines, Bureau of Aircraft Production, 360 Madison Avenue, New York City, (mail) 53 Glen Road, *Jamaica Plain, Mass.*

CALLAHAN, JOHN A. (M) factory manager, Curtiss Aeroplane & Motor Corporation, 65 Churchill Street, *Buffalo, N. Y.*

CAMPBELL, J. HERBERT (M) assistant inspection engineer, Curtiss Aeroplane & Motor Corporation, 65 Churchill Street, *Buffalo, N. Y.* (mail) 22 Woodette Place.

CARRITTE, J. P. (A) general manager, treasurer, McAdamite-Aluminum Co., 57-83 Isabella Avenue, *Detroit, Mich.*

CARSTENS, GEORGE E. (J) chief draftsman, Walker Vehicle Co., 531 West Thirty-ninth Street, *Chicago, Ill.*

CHASE, LEON W. (M) major, Engineering Division, Ordnance Corps, Washington; professor of agricultural engineering, University of Nebraska, University Farm, Lincoln, Neb. (mail) *Ballston, Va.*

CHURCH, EDWIN F., JR. (Aff. Rep.) professor of mechanical engineering, Polytechnic Institute of Brooklyn, 99 Livingston Street, *Brooklyn, N. Y.*

COLE, R. A. (J) chief draftsman, Hayes Wheel Co., *Jackson, Mich.*

COLES, WILFRED G. (M) chief engineer, Madison-Kipp Lubricator Co., *Madison, Wis.* (mail) 421 Sidney Street.

CONNER, JEFFERSON T. (J) specification engineer, Continental Motors Corporation, *Detroit, Mich.* (mail) 346 Hamilton Avenue.

CORTELYOU, JOSEPH S. (A) vice-president, Automobile Trade Directory, 243 West Thirty-ninth Street, New York City, (mail) *Haworth, N. J.*

CREAGER, EMORY F. (J) draftsman, Airplane Engineering Dept., McCook Field, Dayton, Ohio, (mail) R. F. D. No. 1, *Oakland Village, Dayton, Ohio.*

CRITCHLOW, JOHN N. (A) inspection engineer, United Alloy Steel Co., *Canton, Ohio.*

DE LAVANDEYRA, ALBERTO (M) president, chief engineer, Acieral Co. of America, Inc., 26 Cortlandt Street, *New York City.*

DICKEY, CLYDE E. (A) production expert, New York district, Motors Branch, Quartermaster Corps, 469 Fifth Avenue, New York City, (mail) Dickey Steel Co., 233 Broadway, *New York City.*

DORFMÜLLER, ANTON (J) mechanical engineer, S K F Ball Bearing Co., *Hartford, Conn.* (mail) 15 Warrenton Avenue.

DRYSDALE, WILLIAM D. (A) tool designer, Nordyke & Marmon Company, *Indianapolis, Ind.* (mail) 1342 Barth Avenue.

DURNELL, FRANK A. (A) 4029 Broadway, *Indianapolis, Ind.*

EDWARDS, CLYDE L. (A) chief inspector, Dort Motor Car Co., *Flint, Mich.* (mail) 1218 South Saginaw Street.

ENYEART, HARRY L. (J) Auburn Automobile Co., *Auburn, Ind.* (mail) 333 West Eleventh Street.

FAIRMONT GAS ENGINE & RAILWAY MOTOR CAR CO. (Aff.) *Fairmont, Minn.* Representatives: F. E. Wade, president and general manager; W. F. Kasper, mechanical engineer.

FERRIS, HARRY L. (A) service manager, Autocar Sales Co., 533 West Twenty-third Street, *New York City.*

FISHER, CHARLES S. (M) chief engineer, Detroit Accessories Corporation, Gratiot and Fisher streets, *Detroit, Mich.* (mail) 1624 West Grand Boulevard.

FRENCH, W. F. (A) secretary, treasurer and general manager, Louisiana Motor Car Co., Inc., *Shreveport, La.* (mail) 88 Jordan Street.

GANNETT, HERBERT INGALLS (M) mechanical engineer and secretary, Douglas Motors Corporation, *Omaha, Neb.* (mail) 5115 Davenport Street.

GAU, EMIL (J) instructor, Running Motor Department, University of Cincinnati, *Cincinnati, Ohio,* (mail) 2207 Gest Street.

GIERN, JAMES B. (A) president and manager, Giern & Anholtt Tool Works, Inc., Forty-third Street and Aubin Avenue, *Detroit, Mich.*

GILBERT, FREDERICK C. (M) vice-president, Timken-Detroit Axle Co., *Detroit, Mich.* (mail) 162 Virginia Park.

GODDARD, CONRAD G. (M) engineer, 33 East Fiftieth Street, *New York City.*

GOSSETT, P. L. (J) tool department, Premier Motor Corporation, *Indianapolis, Ind.* (mail) 441 North Riley Avenue.

GRAVES, WALTER S. (A) chief engineer, Shotwell-Johnson Co., 712 Ontario Avenue, N., *Minneapolis, Minn.*

HANSON, HENRY L. (J) draftsman, Fergus Motors of America, Inc., 370 Jelliff Avenue, Newark N. J., (mail) 177 Water Street, *Perth Amboy, N. J.*

HAPPEL, ALBERT W. (M) chief draftsman, truck department, Willys Overland Co., *Toledo, Ohio,* (mail) R. F. D. No. 9, Box 128A.

HAUSER, GEORGE H., JR. (J) assistant project engineer, Curtiss Engineering Corporation, Garden City, N. Y. (mail) *East Williston, N. Y.*

HEBNER, ALFRED K. (A) secretary, general manager and mechanical engineer, Bearings Service Co., 733 Cass Avenue, *Detroit, Mich.*

HECKER, RALPH EDWIN (A) traveling mechanical engineer, Norma Co. of America, 1790 Broadway, *New York City.*

HILTON, W. P. (A) certified public accountant and systematizer, 719 Bank of Commerce Building, *Norfolk, Va.*

JACKSON, PHILIP B. (J) designer, Pierce-Arrow Motor Car Co., Elmwood Avenue, Buffalo, N. Y. (mail) 249 Fletcher Street, *Tonawanda, N. Y.*

Karns, W. C. (A) manager, A. M. Karns & Sons, *Everett, Pa.*

Kasper, W. F. (Aff. Rep.) president and general manager, Fairmont Gas Engine & Railway Motor Car Co., *Fairmont, Minn.*

Lansing, J. Twichell (A) second assistant manager, Bijur Motor Appliance Co., River Front and Fifteenth streets, Hoboken, N. J. (mail) 16 Grove Terrace, *Montclair, N. J.*

Lewis, Will I. (A) service manager, Reliance Auto Co., *Birmingham, Ala.*

Lewis, William Stewart (A) manager Detroit office, L. H. Gilmer Co., Tacony, Philadelphia, Pa. (mail) 965 Woodward Avenue, *Detroit, Mich.*

Macaulay, D. L. (A) Sun Life Assurance Co. of Canada, *Montreal, Canada.*

Martell, Leonard R. (A) president and treasurer, Detroit Accessories Corporation, 2021 Gratiot Avenue, *Detroit, Mich.* (mail) 639 Helen Avenue.

Meredith, James Alridge (S. E.) student, 1413 Spencer Avenue, *Marion, Ind.*

Merritt, Ralph N. S. (J) assistant, experimental laboratory, Nordyke & Marmon Co., *Indianapolis, Ind.* (mail) 521 North Riley Avenue.

Miller, Lestere (A) president and designer, Miller Aeroplane & Tractor Co., Dallas, Texas, (mail) P. O. Box 1049, *Dayton, Ohio.*

Morrison, G. Elliott (J) engineer, Breese Aircraft Corporation, Farmingdale, N. Y. (mail) 913 Sterling Place, *Brooklyn, N. Y.*

Morse, William G. (A) factory manager, Austin Mfg. Co., *Harvey, Ill.*

Nelson, John H. (M) research engineer, Wyman-Gordon Co., *Worcester, Mass.*

Nishiyama, F. (M) mechanical engineer, Engineering Department, Tokyo Military Arsenal, *Tokyo, Japan.*

Oxberry, Sydney (A) editor, Class Journal Co., 239 West Thirty-ninth Street, New York City, (mail) 42 Treno Street, *New Rochelle, N. Y.*

Pack, Charles (M) chief chemist and metallurgist, Doehler Die-Casting Co., Court, Ninth and Huntington streets, *Brooklyn, N. Y.*

Parker, George C. (M) chief, engineering department, Cincinnati Ball Crank Co., *Oakley, Cincinnati, Ohio.*

Parker, Worthington F. (A) sales manager, Packard Electric Co., *Warren, Ohio.*

Pendleton, E. R. (M) chief tractor engineer, Engel Aircraft Co., 111 Erie Street, *Niles, Ohio.*

Peterson, C. H. (M) superintendent, Olds Motor Works, *Lansing, Mich.*, (mail) 514 Butler Street.

Polytechnic Institute of Brooklyn (Aff.) 99 Livingston Street, *Brooklyn, N. Y.* Representatives: Edwin F. Church, professor of mechanical engineering.

Price, W. T. (M) president, Price Engine Corporation, 20 South Fifteenth Street, *Philadelphia, Pa.*

Reid, Joseph Boyd (M) engineer and senior inspector airplanes and airplane engines, Bureau of Aircraft Production, 360 Madison Avenue, New York City, (mail) 682 East Second Street, *Brooklyn, N. Y.*

Robbins, Edward A. (A) 39 Cedar Street, *Taunton, Mass.*

Robertson, Malcolm H. (A) automobile accessories department, Marshall-Wells Co., *Duluth, Minn.*

Rowland, George R. (M) supervising engineer, lubricating division, Texas Co., 17 Battery Place, *New York City.*

Ruzicka, John W. (A) proprietor, Motor Electrical Equipment Co., 1251 Michigan Avenue, *Chicago, Ill.*

Sakuyama, James Yeikichi (M) designer, Allison Experimental Co., Speedway, *Indianapolis, Ind.*

Schafer, John V. (J) assistant engineer, Caskey-Dupree Mfg. Co., *Marietta, Ohio,* (mail) 407 Fourth Street.

Scribner, William L. (M) plant engineer, Standard Parts Co., Bock Bearing Division, *Toledo, Ohio,* (mail) Bock Bearing Co.

Schwab, Louis (A) president, Stevens & Co., 375 Broadway, *New York City.*

Schwable, A. G. (A) director of purchases, Curtiss Aeroplane & Motor Corporation, 2000 Elmwood Avenue, *Buffalo, N. Y.*

Smith, John W. (M) manufacturer, 1135 South Fifty-eighth Street, *Philadelphia, Pa.*

Stetter, John M. (A) president and manager, Muncie Cap & Set Screw Co., *Muncie, Ind.*

Steiner, Hubbard W. (M) assistant engineer, Standard Parts Co., Hickox Building, *Cleveland, Ohio.*

Threader, Wilfred G. (A) captain, engineer office and pilot, Bureau of Aircraft Production, Langley Field, *Hampton, Va.*

Tiedmann, Henry (A) service engineer, Splitdorf Electrical Co., 98 Warren Street, *Newark, N. J.*

Topie, Edward G. (A) first lieutenant, supervisor of tools, Ordnance Department, (mail) Locomobile Co., *Bridgeport, Conn.*

Uhl, Henry W. (M) designing engineer, Eisemann Magneto Co., 32 Thirty-third Street, Brooklyn, N. Y. (mail) 107 N. Long Beach Avenue, *Freeport, N. Y.*

Van de Velde, Paul George (M) French High Commission, 65 Broadway, *New York City.*

Van Husan, Corwin (J) instructor in ground aeronautics, Air Service, (mail) 655 E. Jefferson Avenue, *Detroit, Mich.*

Wade, F. E. (Aff. Rep.) mechanical engineer, Fairmont Gas Engine & Railway Motor Car Co., *Fairmont, Minn.*

Wagner, P. C. (J) sales engineer, Blood Brothers Machine Co., *Allegan, Mich.*

Waterman, J. M. (M) assistant chief engineer, Canadian Aeroplanes, Ltd., Dufferin Street and Lappin Avenue, *Toronto, Canada,* (mail) 13 Astley Avenue.

Watkins, Roy Avery (A) vice-president and general manager, Bath Machine Works, Inc., *Bath, N. Y.*

Webster, R. C. (M) works manager, Parrett Tractor Co., Seventeenth Street and Center Avenue, *Chicago Heights, Ill.*

Wells, George B. (M) lieutenant, tire engineer, Engineering Division, Motor Transport Corps, 358 Union Station, Washington, (mail) 111 East Third Street, *Frederick, Md.*

Wendland, Herman J. (A) Dorris Motor Car Co., 4100 Laclede Avenue, *St. Louis, Mo.* (mail) 4103 Maryland Avenue.

White, E. Field (M) 2307 East Jefferson Avenue, *Detroit, Mich.*

White, James A. (M) chief engineer, Harrison Radiator Corporation, *Lockport, N. Y.*

Winter, H. L. (M) assistant inspection engineer, Curtiss Aeroplane & Motor Corporation, 2000 North Elmwood Avenue, *Buffalo, N. Y.* (mail) 290 Summit Avenue.

Wittemann, Paul W. (M) secretary and treasurer, Wittemann-Lewis Aircraft Co., Inc., Lincoln Highway, *Newark, N. J.*

Zeichner, Frank H. (M) planning and tool engineer, Fourth Gun Plant, Poole Engineering & Machine Co., Railroad and Union avenues, *Woodberry, Baltimore, Md.* (mail) 3905 Fernwood Avenue.